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The effect of high exhaust pressures on engine performance and the availability of energy in exhaust gases at cruising conditions

Bliss, Louis K.; Hughes, Joseph W.; Koch, Lincoln; Powell, Lucien C.

Massachusetts Institute of Technology

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EFFECT OF HIGH EXHAUST PRESSURES
ON ENGINE PERFORMANCE AND THE
AVAILABILITY OF ENERGY IN EXHAUST
GASES AT CRUISING CONDITIONS

BY

LOUIS K. BLISS
JOSEPH W. HUGHES
LINCOLN KOCH
AND
LUCIEN C. POWELL

Thesis
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THE EFFECT OF HIGH EXHAUST PRESSURES
ON ENGINE PERFORMANCE AND THE AVAILABILITY OF ENERGY
IN EXHAUST GASES AT CRUISING CONDITIONS

by

Handwritten signature
Ed 28

Lieut. Comdr. Louis K. Bliss, USN
Lieut. Comdr. Joseph W. Hughes, USN
Lieut. Comdr. Lincoln Koch, USN
Lieut. Comdr. Lucien C. Powell, USN

Submitted in Partial Fulfillment of the Requirements for the

Degree of

Master of Science in

Aeronautical Engineering

from the

Massachusetts Institute of Technology

1946

THE CHIEF OF THE BUREAU OF THE ARMY
THE CHIEF OF THE BUREAU OF THE NAVY
THE CHIEF OF THE BUREAU OF THE AIR FORCE

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Submitted in Partial Fulfillment of the Requirements for the

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It is not possible to answer

unpleasantly surprised

Department of Psychology

544

Cambridge, Massachusetts,
3 June 1946.

Professor George W. Swett,
Secretary of the Faculty,
Massachusetts Institute of Technology,
Cambridge, Massachusetts.

Dear Sir:

A thesis entitled "The Effect of High Exhaust Pressures on Engine Performance and the Availability of Energy in Exhaust Gases at Cruising Conditions" is herewith submitted in partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering.

Respectfully,

the use of a number of other in addition to the
also included in the list of the requirements
which in Japan are of great importance" is the
statement on the Japanese and the availability of
A thesis entitled "The Effect of the Japanese
classification.

John B. King

— 1900 —

John Smith
John Smith, Esq.

1870

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ACKNOWLEDGMENTS

Taylor, C. Fayette, Massachusetts Institute of Technology.

Leary, W. A., Massachusetts Institute of Technology.

Livengood, J. C., Massachusetts Institute of Technology.

General Electric Company, Schenectady, New York.

CONTENTS

Taylor, G. W., *Investigations in the History of Technology*.

Levy, W. A., *Investigations in the History of Technology*.

Livermore, F. W., *Investigations in the History of Technology*.

General Electric Company, *General Electric*, New York.

SYMBOLS

A	Area, in square inches
B	Ratio of diameter of orifice to diameter of pipe
BHP	Brake horsepower
BL	Brake load, inches of mercury
e	Volumetric efficiency
F	F/A = fuel-air ratio
H	Orifice pressure drop, inches of water
h_1	Gage manifold pressure, inches of mercury
h_e	Gage exhaust pressure, inches of mercury
I_{ex}	Exciter current, amperes
K	Orifice coefficient
l	Stroke, in inches
L	Brake arm length, in inches
m.e.p.	Mean effective pressure, p.s.i.
BMEP	Brake mean effective pressure, p.s.i.
FMEP	Friction mean effective pressure, p.s.i.
IMEP	Indicated mean effective pressure, p.s.i.
PMEP	Pumping mean effective pressure, p.s.i.
N	Revolutions per minute
P	Pressure ahead of orifice, inches of mercury, equal to P_1 plus corrected barometric pressure.
P_1	Gage pressure ahead of orifice, inches of mercury
P_e	Exhaust pressure, inches of mercury
P_i	Inlet (manifold) pressure, inches of mercury

SYMBOLS (continued)

Rot.	Rotameter reading
S.A.	Spark advance, in degrees before top dead center
T	Temperature of air entering orifice, in deg. R.
T_1	Inlet (manifold) temperature, deg. R.
T_1 }	Temperature measured in calorimeter, deg. F.
T_2 }	
T_3 }	
T_4	Temperature in exhaust pipe, deg. F.
T_e	Exhaust temperature, deg. F, used in calculation of turbine work
W_A	Mass rate of airflow, pounds per hour
W_F	Mass rate of fuel flow, pounds per hour
W_c	Compressor work, Btu/lb. air
W_t	Turbine work, Btu/lb. air
W_e	Engine brake work, Btu/lb. air
Y	Expansion factor

SUMMARY

The combination of two types of power plants to give optimum performance is one solution to demands for ever increasing power for high speed aircraft. A compressor, engine, and turbine geared to the same shaft is one such combination. Any prediction of the performance of such a system depends upon the experimental determination of the power output of the engine.

The purpose of this investigation was to determine the output of the engine at high exhaust pressures and the energy of the exhaust gases at these pressures. Values at full power conditions were obtained at the Sloss Laboratory in May 1944. (Ref. 1.) The present investigation involved the determination of these variables at cruise conditions.

A Lycoming single cylinder engine was used for the investigation. The engine was operated at a piston speed of 2100 feet per minute, with a fuel-air ratio of .065, spark advance of 28 degrees, manifold inlet temperature of 140°F, and a valve overlap of 40 degrees. The cylinder head temperature, oil pressure, and oil temperature were maintained constant at normal values.

Exhaust pressure was varied from 30 to 60 inches of mercury in 10-inch increments, and the manifold pressure from 30 to 50 inches of mercury in the same increments. Sufficient data was recorded to plot a map of the region under investigation.

SYNOPSIS

The combustion of two types of power plants is also optimum performance is one subject to be considered for their increasing power for high speed aircraft. A comparison, engine, and engine tested to the same effect in one case a comparison. The prediction of the performance of each a system depends upon the experimental determination of the power output of the engine.

The purpose of this investigation was to determine the output of the engine at high engine pressures and the energy of the exhaust gases at these pressures. Values at full power conditions were obtained at the Army Laboratory in May 1944. (Ref. 1.) The present investigation involved the determination of these variables at cruise conditions.

A four-cylinder single cylinder engine was used for the investigation. The engine was operated at a constant speed of 2100 rpm per minute, with a fuel-air ratio of 0.06, a pressure of 28 lb per sq in, and a temperature of 140°F. and a valve opening of 40 degrees. The cylinder head temperature, oil pressure, and oil temperature were maintained constant at normal values.

Exhaust pressure was varied from 30 to 60 inches of mercury in 10-inch increments, and the exhaust pressure from 30 to 50 inches of mercury in the same increments. The exhaust data was recorded to plot a map of the region under investigation.

The energy of the exhaust gases was obtained by measurement of the exhaust temperatures.

It can be concluded as a result of this investigation that:

1. In the range investigated, the mass rate of airflow decreases linearly with increase in exhaust pressure.
2. Volumetric efficiency falls off linearly with increase in exhaust pressure.
3. Brake Horsepower decreases almost linearly with increase in exhaust pressure.
4. BMEP and IMEP decrease linearly with increase in exhaust pressure.
5. Indicated Horsepower increases linearly with mass airflow.
6. Mechanical FMEP and FMEP increase linearly with increase in exhaust pressure.
7. FMEP decreases linearly and mechanical FMEP increases with increase in inlet pressure.
8. Brake specific air consumption increases with increase in exhaust pressure, the rate of increase becoming greater with larger values of exhaust pressure.
9. Exhaust temperature increases linearly with increase in exhaust pressure.

The energy of the exhaust gases was obtained by means of

one of the exhaust gases.

It can be considered as a result of this investigation

that

1. In the case investigated, the same rate of

exhaust gases increases linearly with increase in exhaust

pressure.

2. Volumetric efficiency falls off linearly with

increase in exhaust pressure.

3. Brake horsepower decreases almost linearly with

increase in exhaust pressure.

4. MEP and BHP decrease linearly with increase in

exhaust pressure.

5. Indicated horsepower decreases linearly with mean

effective pressure.

6. Mechanical efficiency and BHP increase linearly with

increase in exhaust pressure.

7. MEP decreases linearly and volumetric efficiency falls

exponentially with increase in exhaust pressure.

8. Brake specific fuel consumption increases with

increase in exhaust pressure. The rate of increase

becoming greater when larger values of exhaust pres-

sure.

9. Exhaust temperature increases linearly with

increase in exhaust pressure.

10. Optimum engine p_0/p_1 for the CET system increases with altitude and occurs between the values .75 and 1.0.

11. Net mean effective pressure of the CET system is slightly higher at cruising conditions than at full power conditions.

12. Net specific fuel consumption for the CET system is lower at cruising conditions than at full power conditions.

13. The turbine output in a CET system is more likely to be limited by the turbine blading material than by the energy available in the exhaust gases.

This investigation was made at the Sloan Automotive Laboratory at Massachusetts Institute of Technology, Cambridge, Massachusetts, in April-May, 1946.

10. The first stage of the investigation was to determine the effect of the various factors on the rate of reaction. The results are shown in Table I.

11. The second stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table II.

12. The third stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table III.

13. The fourth stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table IV.

14. The fifth stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table V.

15. The sixth stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table VI.

16. The seventh stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table VII.

17. The eighth stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table VIII.

18. The ninth stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table IX.

19. The tenth stage was to determine the effect of the various factors on the rate of reaction. The results are shown in Table X.

INTRODUCTION

The wartime demands of the military services for aircraft capable of operating at higher altitudes and at greater speeds stimulated extensive investigation into the development of new power plants with ever higher output. The use of different types of power plants in combination to give optimum performance is a solution that offers unevaluated possibilities. The compressor-engine-turbine system is one such combination. Any prediction of the performance of such a system depends, because of the number of variables involved, upon the experimental determination of the power output of the internal combustion engine under conditions of high exhaust pressure and a measurement of the residual energy of the exhaust gases. (Ref. 2.)

The determination of these factors at full power operation was made at the Sloan Automotive Laboratory at Massachusetts Institute of Technology in April 1944. (Ref. 1.) The present investigation is a continuation of that work in the determination of these factors at cruising conditions. This work was done in the Sloan Laboratory at Massachusetts Institute of Technology in April and May, 1946.

INTRODUCTION

The entire demand of the military services for air-

craft capable of operating at higher altitudes and at
greater speeds stimulated extensive investigation into the
development of new power plants with ever higher output.

The use of different types of power plants in combination
to give optimum performance in a particular class of aircraft
evaluated possibilities. The turbo-propeller-engine-turbine

system is one such combination. The production of the
performance of such a system depends, because of the number
of variables involved, upon the experimental determination
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conditions of high exhaust pressure and a substantial ex-

haust energy of the exhaust gases. (Ref. 1.)
The investigation of these factors at full power oper-

ation was made at the Glenn Aircraft Laboratory at
Massachusetts Institute of Technology in April 1944. (Ref. 2.)
The present investigation is a continuation of that work in
the determination of these factors at engine conditions.
This work was done in the Glenn Laboratory at Massachusetts
Institute of Technology in April and May, 1944.

DESCRIPTION OF APPARATUS

The set-up and arrangement of the apparatus used in this investigation is shown by the photographs, Figs. A, B and C, and by a diagrammatic sketch in Fig. D.

The engine was a single cylinder Lycoming on a Universal crankcase. It was liquid cooled, had dual spark ignition, a compression ratio of 6, a bore of 5.25 inches, and a stroke of 6.25 inches. It had single inlet and exhaust valves with an overlap of 40 degrees. The valve timing diagram is shown in Fig. E. The engine drove, besides the dynamometer, only the valve gear, the ignition breaker points, the tachometer, and the dynamometer exciter. The engine accessories, i.e., fuel, oil and water pumps, were driven by a three phase induction motor which also served as a starter.

The power output was absorbed by a Reliance Eddy Current Dynamometer. The brake load was measured by a hydraulic scale sometimes referred to as a torque cell. Brake load readings were obtained on a mercury manometer. The load was varied by varying the field current of the exciter which was cascaded through a motor generator set to load the dynamometer.

The Speed was controlled by varying the load. It was measured closely by a standard tachometer and checked by means of a 60-cycle stroboscopic light directed on a pattern disc on the flywheel.

DESCRIPTION OF APPARATUS

The motor and arrangement of the apparatus used in this investigation is shown by the photographs, Figs. 1, 2 and 3, and by a schematic diagram in Fig. 4.

The engine was a single cylinder operating on a Diesel cycle. It was liquid cooled, had dual valves, a compression ratio of 16, a bore of 3.5 inches, and a stroke of 6.5 inches. It had electric inlet and exhaust valves with an opening of 1.5 inches. The valve timing diagram is shown in Fig. 5. The engine drove, besides the dynamometer, only the valve gear, the fuel pump, the water pump, the compressor, and the dynamometer. The engine was driven by a three phase induction motor which also served as a generator.

The power output was measured by a Bellows type dynamometer. The brake load was measured by a spring scale mounted on a torque coil. Brake load readings were obtained on a spring scale. The load was varied by varying the field current of the motor which was connected through a motor generator set to load the dynamometer.

The speed was controlled by varying the load. It was measured electrically by a standard tachometer and checked by means of a 50-cycle synchronous light alternator on a potentiometer on the flywheel.

The spark was controlled remotely from the instrument panel. The advance was measured by the conventional type neon spark disc.

The engine inlet system consisted of a combination surge and vaporizing tank. Temperature control of the inlet air was obtained by supplying either low pressure steam or cooling water to the surge tank jacket. Intake air was supplied by laboratory compressors, the flow being controlled remotely from the control panel by a Minneapolis-Honeywell gate valve which bled to the atmosphere. The air was metered in the induction line by means of a sharp-edged orifice installed in accordance with ASME specifications. (Refs. 3 and 4.) The pressure drop across the orifice was measured on a differential water manometer. The pressure before the orifice and in the surge tank was measured by mercury manometers. The temperature at the corresponding points was measured by Bureau of Standards iron-constantan thermocouples and a Tagliabu potentimeter.

One hundred octane fuel was supplied from the laboratory system. The pressure was maintained constant by the externally driven fuel pump vented to the intake system through a back-pressure diaphragm arrangement. The rate of fuel flow was controlled by a valve on the control panel and was measured by a Fischer and Porter Stahl-Vis Rotameter.

The cooling water and oil were circulated in closed systems by means of externally driven pumps. Temperature control

The pump was controlled locally from the instrument

panel. The pressure was measured by the conventional type

manometer also.

The engine inlet system consisted of a combination

of a surge tank and a surge tank. The surge tank was

fed air was supplied by supplying air from the surge tank

or cooling water to the surge tank. Inlet air was

supplied by laboratory compressors, the flow being controlled

remotely from the control panel by a remotely-controlled

gate valve which acted on the atmosphere. The air was

in the instrument line by means of a sharp-edged orifice in-

stalled in accordance with ASME specifications. (See 1) and

4.) The pressure drop across the orifice was measured on a

differential water manometer. The pressure before the orifice

line and in the surge tank was measured by mercury manometers.

The temperature at the corresponding points was measured by

thermocouples of standard type-constantan thermocouples and a

thermocouple potentiometer.

The heated gases from the laboratory

space. The pressure was maintained constant by the ex-

actly driven fuel gas valve in the instrument which a

back-pressure diaphragm arrangement. The rate of fuel flow

was controlled by a valve on the control panel and was

measured by a Fischer and Porter Model-Via Rotameter.

The cooling water and oil were circulated in closed sys-

tems by means of externally driven pumps. Temperature control

of these systems was by means of heat exchangers that could be supplied with either cooling water or low-pressure steam through valves located at the control panel.

The exhaust gases were discharged through a short coupling into an exhaust calorimeter. From the calorimeter, the exhaust gases were led to a water-jacketed surge tank where they were partially cooled and then exhausted through a Minneapolis-Honeywell gate valve which was used to control the exhaust pressure. The exhaust pressure was measured by means of a mercury manometer from a tap at the exhaust calorimeter.

The exhaust calorimeter was that designed and used by the authors of Ref. 1. A scale drawing of the calorimeter is shown in Fig. F. Essentially, it is a shell within a shell arrangement fitted with baffles to retard and mix the exhaust gases. The outer shell was provided with bleeds by means of which some of the hot gases could be passed around the inner shells in order to reduce radiation losses from the center of the calorimeter where the thermocouples were located. For this investigation the only modification of the apparatus was to increase the size of these bleeds. The valves in the bleed lines were provided to control the amount of gas flowing around the inner shells. Three thermocouples were used and consisted of porcelain shielded Chromel-Alumel junctions. The temperatures given by these couples were read on a Leeds and Northrup millivoltmeter and were converted to $^{\circ}\text{F}$ by use of

of these systems was the amount of heat transferred from the
be supplied with steam cooling water by low-pressure steam
through valves located at the control panel.

The exhaust gases were directed through a duct leading
into an exhaust collector. From the collector, the ex-
haust gases were led to a water-heated duct and where they
were partially cooled and then exhausted through a chimney.
Exhaust gas water which was used to control the exhaust
pressure. The exhaust pressure was measured by means of a
mercury manometer from a tap at the exhaust collector.
The exhaust collector was first designed and used by
the authors of Ref. 1. A scale drawing of the collector is
shown in Fig. 2. Essentially, it is a shell within a shell
arrangement fitted with baffles to retard and stir the exhaust
gases. The outer shell was provided with flange by means of
which some of the hot gases could be passed around the lower
shell in order to reduce radiation losses from the center of
the collector where the thermocouples were located. For
this investigation the only modification of the system was
to increase the size of these flanges. The valve in the flange
lines were provided to control the amount of gas flowing
around the inner shell. These thermocouples were used not
consisted of porcelain-enclosed Chromel-Alumel junctions. The
temperatures given by these couples were read on a scale and
hot-wire millivoltmeter and were converted to $^{\circ}\text{F}$ by use of

Tables in Ref. 5. A fourth thermocouple, designed and manufactured by the General Electric Company for the measurement of high temperature, high velocity gases (Ref. 6) was inserted in the cylinder exhaust stack ahead of the calorimeter.

The M.I.T. High Speed Indicator was used to obtain records of cylinder pressure versus crank angle. The M.I.T. transfer table was used to obtain pressure-volume diagrams therefrom.

A Dumont Cathode Ray Oscilloscope was used in conjunction with a Sperry Magneto striction vibration pickup to detect any possible detonation.

PROCEDURE

The information sought divided the experimental work into three natural subdivisions. First, that of measuring the brake horsepower, the mass air flow, and the corresponding volumetric efficiency; second, the determination of the IHP or IMEP and m.e.p. by means of indicator cards; and third, that of measuring the exhaust temperature, all in relation to varying inlet and exhaust pressures.

The first and third quantities were determined by one set of runs using dual ignition. The second quantity was necessarily obtained by a second series of runs because one spark plug had to be removed in order to insert an indicator plug to obtain the cylinder pressure for the indicator. All runs were made with a concentrated effort to maintain uniformity and to duplicate the running conditions of Ref. 1 except that cruising r.p.m. and fuel-air ratio were used instead of full power r.p.m. and fuel-air ratio.

All runs were made maintaining engine operating variables as near the following values as possible:

Piston speed (Cruising) = 2100 ft./min. (2025 rpm)

F/A (Cruising) = $0.065 \pm .0010$

T_1 = $140 \pm 1^\circ\text{F.}$

Adjustments were made to give

S.A. = 25° for dual ignition

= 35° for single plug operation (used during taking indicator cards.)

RESULTS

The information herein is presented with
 into three separate sections. First, that of measuring
 the engine horsepower, the mean air flow, and the horsepower-
 per volumetric efficiency; second, the determination of the
 H.P. or B.M.P. and M.E.P. by means of indicator cards; and
 third, that of measuring the exhaust temperature, all in
 relation to varying load and exhaust pressure.

The first and third quantities were determined by means
 of two using dual indicator. The second quantity was
 determined by a second series of runs because the
 engine had to be removed in order to insert an indicator
 plug to obtain the cylinder pressure for the indicator. All
 runs were made with a concentrated effort to maintain vol-
 ume and to duplicate the running conditions of test 1
 except that instead of 1.5 and 100-1/2 psi were used in-
 stead of full power 1.5 and 100-1/2 psi.

All runs were made maintaining engine operating con-
 dition as near the following values as possible:

1.5 (volts)	2.00 ± .005
100-1/2 (psi)	1.0 ± .05

Adjustments were made to give

1.5 for dual indicator	2.00 ± .005
100-1/2 for single plug oper-	1.0 ± .05
Mean (read during	
taking indicator cards.)	

The operating temperatures and pressures were maintained within the following limits:

Oil Pressure	58 - 70 p.s.i.
Inlet Oil Temperature	150°F.
Cylinder Temperature	175 - 185°F.

The first series of runs was made under the following varying conditions:

<u>Inlet Pressure, "Hg.</u>	<u>Exhaust Pressure, "Hg.</u>
30	30
30	40
30	50
30	60
40	30
40	40
40	50
40	60
50	30
50	40
50	50
50	60

The second series of runs was made under the following varying conditions:

<u>Inlet Pressure, "Hg.</u>	<u>Exhaust Pressure, "Hg.</u>
30	40
40	30
40	40
40	50
40	60
50	40

Several familiarization runs were made to determine the best operating procedure. The ranges of values that might

The operating temperatures and pressures were maintained

within the following limits:

Oil Temperature 95 - 100 F.S.I.

Water Oil Temperature 120 F.

Cylinder Temperature 175 - 185 F.

The first series of runs was made under the following

varying conditions:

Injection Pressure, "PSI.

30	30
35	35
40	40
45	45
50	50
55	55
60	60
65	65
70	70
75	75
80	80
85	85
90	90

The second series of runs was made under the following

varying conditions:

Injection Pressure, "PSI.

30	30
35	35
40	40
45	45
50	50
55	55
60	60
65	65
70	70
75	75
80	80
85	85
90	90

Several fertilization runs were made to determine the

best operating procedure. The ranges of values used were

be expected were determined, and calibrations, adjustments, and zero or tare readings were all determined during this series of familiarization runs.

Two complete series of record runs were made for the first set of conditions. Mass airflow, brake horsepower, and volumetric efficiency were computed from readings of the orifice manometers, thermocouples, and the brake load manometer. Exhaust temperatures were obtained from the calorimeter and exhaust stack thermocouples.

A single series of runs was made for the second set of conditions. Pressure versus crank angle curves were obtained with the M.I.T. High Speed Indicator. Both light (5 lbs/in) and heavy (150 lbs/in) springs were used in order to obtain an accurate record of the effect of exhaust pressure on the PMEP as well as its effect on the IMEP.

to control were dissolved, and collected, and removed, and then we have worked with all specimens during this series of familiarization runs.

The complete series of second runs were made for the first set of conditions. Then after, from subsequent, and volumetric titration with corrected from readings at the other end of the range, and the curve for the temperature were obtained from the colorimeter and plotted on the graph.

A single series of runs was made for the second set of conditions. From the various other series were obtained with the N.I.T. and good results. With 10% (10%) and 20% (10%) series were used in order to obtain an accurate record of the effect of exposure to the pump as well as its effect on the N.I.T.

The results of the runs were plotted on the graph and the curves were obtained.

Table I. Results of runs.

Run	1	2	3	4	5	6	7	8	9	10
1	10	10	10	10	10	10	10	10	10	10
2	10	10	10	10	10	10	10	10	10	10
3	10	10	10	10	10	10	10	10	10	10
4	10	10	10	10	10	10	10	10	10	10
5	10	10	10	10	10	10	10	10	10	10
6	10	10	10	10	10	10	10	10	10	10
7	10	10	10	10	10	10	10	10	10	10
8	10	10	10	10	10	10	10	10	10	10
9	10	10	10	10	10	10	10	10	10	10
10	10	10	10	10	10	10	10	10	10	10

The results of the runs were plotted on the graph and the curves were obtained.

DISCUSSION OF RESULTS

This investigation paralleled rather closely the work described in Ref. 1. An effort was made to avoid deviation from the procedure and methods outlined therein, in order that direct comparisons could be made between results obtained for the full power conditions and those at cruising power.

In the time elapsed between the initiation of that work and the completion of this, other investigators have done extensive research on the effect of high exhaust pressures on the performance of internal combustion engines. (Ref. 7.) The results reported are more general and greater in scope than could be obtained in this limited work. Further, certain recent refinements and developments in obtaining temperature measurements of high temperature, high velocity gases have been reported, which indicate a more satisfactory means of obtaining the exhaust temperature. (Refs. 6, 8, 9.) Nevertheless, it is felt that the results obtained can be used as a basis for more extensive research in this field.

The experimental results and certain computed values obtained therefrom are shown in Table I. All results are shown by means of curves in Figs. 1 - 22. The original data sheets for all runs, pressure-crank angle diagrams and calibration curves are on file in the Sloan Automotive Laboratory at Massachusetts Institute of Technology.

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of the Institute of Technology.

These curves are on the same logarithmic scale for all cases, pressure-tank angle diagrams and all-
shown by means of curves in Fig. 1 + 2. The original data
obtained therefore are shown in Table I. All results are
The experimental results are shown in Table I. All results are
used as a basis for some extensive research in this field.
Nevertheless, it is felt that the results obtained can be
means of obtaining the exhaust temperature. (Data 6, 8, 9.)
These have been reported, which indicate a more satisfactory
pressure measurements of high temperature, high velocity
also recent refinements and developments in obtaining tem-
peratures are shown in this latest work. Further, con-
The results reported are more correct and greater in scope
on the performance of internal combustion engines. (Data 7.)
associated research on the effect of high pressure processes
and the completion of this, other investigations have been
In the time elapsed between the initiation of this work
power.

The effect of varying inlet and exhaust pressures on engine performance can most easily be reviewed by noting their effect upon (1) Mass rate of airflow; (2) Volumetric efficiency; (3) Mean effective pressures; (4) Brake horsepower and brake specific fuel consumption; and (5) Exhaust temperatures.

MASS RATE OF AIRFLOW:

Fig. 1 shows the effect of varying exhaust pressure on the mass rate of airflow. It is seen that, over the range investigated, for any given inlet pressure there is an almost linear reduction of mass airflow with increasing exhaust pressure. Further, the rate of variation is constant for all inlet pressures except at 50" hg. inlet pressure and low exhaust pressure. This effect is probably due to a slight drop in volumetric efficiency at higher mass rate of airflow.

The condition of inlet pressure of 30" Hg. and exhaust pressure of 60" Hg. was unstable, since the brake load required was the minimum load which could be set on the brake. All readings taken under this condition were regarded as potentially in error, and due account was taken of this in subsequent discussion.

VOLUMETRIC EFFICIENCY:

Fig. 2 shows the variation of volumetric efficiency with exhaust pressure. As would be expected from Fig. 1, it is seen that volumetric efficiency decreases linearly with increasing

exhaust pressure except at 50" Hg. inlet pressures and low exhaust pressures. At this point the maximum volumetric efficiency had probably been approached.

In Fig. 3 a plot is shown of volumetric efficiency vs. ratio of exhaust to inlet pressure (P_e/P_i). This curve is found to agree closely with the theoretical relationship of volumetric efficiency as a function of compression ratio and ratio of exhaust to inlet pressure, given in Ref. 10. Variation from this theoretical curve is probably due to valve overlap, which appears to be slightly excessive for the piston speed used.

The effect of variation of exhaust and inlet pressure on volumetric efficiency is illustrated in its relationship to the pumping loops of indicator diagrams obtained during the second set of runs. It is seen in Fig. 14 that, at a given inlet pressure, the size of the pumping loop increases with exhaust pressure and similarly, at a given exhaust pressure, the pumping loop decreases with an increase of inlet pressure.

MEAN EFFECTIVE PRESSURES:

The effect of varying inlet and exhaust pressures on indicated and pumping mean effective pressures was obtained from an analysis of indicator cards. These cards were obtained by converting the pressure-crank angle diagrams made with the M.I.T. High Speed Indicator to pressure-volume diagrams with the aid of the M.I.T. transfer table.

exhaust pressure curves at 50° F. inlet pressure and low
exhaust pressure. It also shows the maximum volume of
liquid gas probably used.
In Fig. 3 a plot is shown of volume of liquid gas
ratio of exhaust to inlet pressure (P_2/P_1). This curve is
typical of gases which with the associated relationship of
volume of liquid gas as a function of expansion ratio and
ratio of exhaust to inlet pressure, given in Fig. 1. It
appears from this theoretical curve is probably due to ratio
overlap, which appears to be slightly excessive for the given
exhaust ratio.

The effect of variation of exhaust and inlet pressure on
volume of liquid gas is illustrated in its relationship to
the pumping ratio at constant distance obtained under the
exhaust and inlet. It is seen in Fig. 1 that, as a given
inlet pressure, the size of the pumping loop increases with
exhaust pressure and slightly, as a given exhaust pressure,
the pumping loop decreases with an increase of inlet pressure.

THE EFFECT OF VARYING INLET AND EXHAUST PRESSURE ON 20-

The effect of varying inlet and exhaust pressure on 20-
diameter and pumping ratio effective pressure was obtained from
an analysis of indicated data. These data were obtained by
conducting the pressure-volume analysis with the
E.I.V. High Speed Indicator in pressure-volume diagram with
the aid of the E.I.V. indicator table.

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The effects of varying exhaust pressure at a constant inlet pressure of 40" Hg. are shown by the superposition of heavy spring (150 lbs/in) and light spring (5 lbs/in) cards shown in Figs. 10 and 11. It is seen that the indicated mean effective pressure, as measured by the area of the pressure-volume diagram, decreases with increasing exhaust pressure. (Fig. 10.) The pumping mean effective pressures, as measured by the areas of the pumping loops, similarly follow expected trends, the areas increasing with increasing exhaust pressure. (Fig. 11.)

The effects of varying inlet pressure at a constant exhaust pressure of 40" Hg. are shown by the superposition of the heavy and light spring cards in Figs. 12 and 13. As in the case of variable exhaust pressures, values of IMEP and PMP obtained from these cards are in agreement with those which would be expected.

These cards were all necessarily obtained with single plug ignition, because the indicator unit was placed in the hole normally occupied by the other plug, and were made at a spark advance of 35° which had previously been determined as best power spark advance for single plug operation. It should be noted that statements based on quantitative values are necessarily subject to limitations due to variations resulting from single plug operation. However, it is probable that any error is small as volumetric efficiency is essentially the same at similar conditions, as seen in Fig. 2.

The effects of varying inlet pressure at a constant inlet pressure of 60" Hg. are shown in the upper portion of Figure 1 (a) and (b) and also in Figure 1 (c) and (d). It is seen that the inlet pressure has a marked effect on the rate of reaction, as indicated by the change in the rate of reaction with inlet pressure. The reaction rate increases with inlet pressure, as shown in Figure 1 (a) and (b). The reaction rate also increases with inlet pressure, as shown in Figure 1 (c) and (d). The reaction rate is also affected by the inlet temperature, as shown in Figure 1 (e) and (f). The reaction rate increases with inlet temperature, as shown in Figure 1 (e) and (f). The reaction rate is also affected by the inlet concentration, as shown in Figure 1 (g) and (h). The reaction rate increases with inlet concentration, as shown in Figure 1 (g) and (h). The reaction rate is also affected by the inlet pressure, as shown in Figure 1 (i) and (j). The reaction rate increases with inlet pressure, as shown in Figure 1 (i) and (j). The reaction rate is also affected by the inlet temperature, as shown in Figure 1 (k) and (l). The reaction rate increases with inlet temperature, as shown in Figure 1 (k) and (l). The reaction rate is also affected by the inlet concentration, as shown in Figure 1 (m) and (n). The reaction rate increases with inlet concentration, as shown in Figure 1 (m) and (n). The reaction rate is also affected by the inlet pressure, as shown in Figure 1 (o) and (p). The reaction rate increases with inlet pressure, as shown in Figure 1 (o) and (p). The reaction rate is also affected by the inlet temperature, as shown in Figure 1 (q) and (r). The reaction rate increases with inlet temperature, as shown in Figure 1 (q) and (r). The reaction rate is also affected by the inlet concentration, as shown in Figure 1 (s) and (t). The reaction rate increases with inlet concentration, as shown in Figure 1 (s) and (t). The reaction rate is also affected by the inlet pressure, as shown in Figure 1 (u) and (v). The reaction rate increases with inlet pressure, as shown in Figure 1 (u) and (v). The reaction rate is also affected by the inlet temperature, as shown in Figure 1 (w) and (x). The reaction rate increases with inlet temperature, as shown in Figure 1 (w) and (x). The reaction rate is also affected by the inlet concentration, as shown in Figure 1 (y) and (z). The reaction rate increases with inlet concentration, as shown in Figure 1 (y) and (z).

The BMEP was calculated from the known power output of the engine. As mentioned above, IMEP and FMEP were obtained from the area of the indicator diagrams. In Fig. 4, the BMEP at varying inlet conditions is plotted against exhaust pressure. It is seen that the BMEP decreases linearly with increasing exhaust pressure and at approximately the same rate for each inlet pressure. The IMEP at 40" Hg. inlet pressure is shown on the same figure. The similarity of the slopes of the BMEP curves was such that construction of parallel curves through the determined points of IMEP at 30" and 50" Hg. seemed logical. That this assumed relationship between IMEP and exhaust pressure at these inlet pressures is substantially correct is shown by a study of Fig. 5, in which it is seen that the indicated horsepower varies directly with mass airflow.

Fig. 6 shows a breakdown of mean effective pressures at an inlet pressure of 40" Hg. As discussed previously, the IMEP and BMEP decrease and the FMEP increases with increasing exhaust pressure. Assuming that IMEP is the sum of BMEP, FMEP and mechanical FMEP, the curve of mechanical FMEP obtained by differences increases slowly with exhaust pressure.

Fig. 7 shows a similar set of curves in which the exhaust pressure is held constant. FMEP decreases with increasing inlet pressure, substantiating the fact that mechanical FMEP increases with increasing inlet pressures.

The H₂O was collected from the power output

of the engine. As indicated above, H₂O was not

obtained from the area of the indicator diagram. In

Fig. 1, the H₂O at varying inlet conditions is shown

against exhaust pressure. It is seen that the H₂O at

exhaust pressure is increasing rapidly with pressure and at

approximately the same rate for both inlet pressures. The

H₂O at 40° K. inlet pressure is shown as the same figure.

The quantity of the amount of the H₂O at 40° K.

that consists of partial vapor through the determined

points of H₂O at 40° K. and 50° K. is shown in Fig. 1.

This amount relationship between H₂O and exhaust pressure

of these inlet pressures is satisfactorily shown in Fig.

by a study of Fig. 2, in which it is seen that the indicated

percentage varies directly with inlet air.

Fig. 3 shows a comparison of mean effective pressure

at an inlet pressure of 40° K. as discussed previously,

the H₂O and H₂O at 40° K. and the H₂O at 50° K. is the

exhaust exhaust pressure. Assuming that H₂O is the

of H₂O, H₂O and mechanical H₂O, the curve of mechanical

H₂O obtained by difference between H₂O and H₂O at 40° K.

pressure.

Fig. 4 shows a similar set of curves in which the exhaust

pressure is held constant. H₂O decreases with increasing

inlet pressure, substantiating the fact that mechanical H₂O

increases with increasing inlet pressure.

BRAKE HORSEPOWER AND BRAKE SPECIFIC AIR CONSUMPTION:

Curves of BHP and BSAC are included in order that direct comparisons can be made with values of these quantities obtained at full power operation on this same test engine and discussed in Ref. 1. However, since both piston speed and fuel air ratio were varied in making measurements at cruise conditions, only a qualitative comparison seems justified.

Fig. 8 shows variation of BHP with exhaust pressure. As in reference 1 for the condition at full power, there is an approximate linear decrease of BHP with increasing back pressure. Similarly, curves of BSAC vs. exhaust pressure, shown in Fig. 9, follow the same order of increasing BSAC with increasing exhaust pressure.

EXHAUST TEMPERATURES:

The determination of the energy of the exhaust gases poses a difficult problem. It can be obtained by making heat measurements using a heat exchanger or by measurement of the exhaust gas temperature. Its determination by measurement of the exhaust temperature seems the easier, but this method has certain attendant difficulties in the case of a single-cylinder engine. In this case there is a very unsteady flow of the hot exhaust gas, the temperature of which varies widely. This variation from initial opening to final closing of the exhaust valve, in the range of operation of this investigation, was computed to be about 800°F. (Ref. 10.)

THE EFFECT OF TEMPERATURE ON THE RATE OF REACTION

Curves of $\log k$ and $\log k_0$ are plotted in order that direct comparison can be made with values of $\log k_0$ obtained from this work. The curves are shown in Fig. 1. It is seen that the curves are nearly parallel and that the rate of reaction is nearly constant at higher temperatures, only a qualitative comparison being justified.

Fig. 2 shows variation of $\log k$ with constant pressure. It is seen that for the reaction at 100°C., there is an approximate linear decrease of $\log k$ with increasing pressure. Similarly, curves of $\log k_0$ vs. constant pressure, shown in Fig. 3, follow the same order of increasing $\log k$ with increasing pressure.

THEORETICAL CONSIDERATIONS

The determination of the order of the reaction is a difficult problem. It can be obtained by using the method of initial rates or by using the method of half-lives. The method of initial rates is more accurate than the method of half-lives. In this case there is a very marked flow of the reaction rate, the temperature of which varies slightly. This variation from initial equilibrium to final equilibrium of the reaction rate, in the range of operation of this investigation, was assumed to be about

The exhaust calorimeter, as described in Ref. 1 and used in this investigation, was designed as a combination surge tank and mixing chamber. It was felt that measurement of the gas temperature after being mixed and brought substantially to rest in the calorimeter would give a valid measurement of its total energy.

Ref. 1 indicates that no constant temperature could be measured but that there was a wide variation in temperatures within the calorimeter at three points, each located successively further from the exhaust valve. This variation was probably due to radiation losses because the temperatures decreased with distance from exhaust port. It was suggested in Ref. 1 that some of the hot exhaust gases be passed between the calorimeter shells, that form the calorimeter walls, to maintain the wall temperature constant and thereby minimize radiation losses. This proved unsatisfactory as is shown qualitatively in Fig. 15, in which it may be seen that indicated temperatures fell off, rather than all three thermocouples approaching a single equilibrium temperature. Though the basic idea of the calorimeter may be sound, a redesign and refinement is necessary before valid and reproducible results can be obtained by its use.

In order, however, to have some common ground for comparison with results obtained in Ref. 1, the exhaust temperatures measured in the exhaust calorimeter without bleed were used. Readings of the three thermocouples showed the

The present experiment, as described in Part I and used in this investigation, was designed on a combination of the two foregoing types being equal and brought substantially to rest in the experiment would give a valid measurement of the total energy.

Part I indicated that no constant temperature could be maintained but that there was a wide variation in temperatures within the calorimeter at three points, each located approximately 1/3 of the way from the exposed side. This variation was probably due to radiation losses between the thermometer and the vessel from which heat was lost. It was suggested in Part I that some of the heat should come from the bottom of the calorimeter rather than the sides, and that the calorimeter be made to maintain the wall temperature constant and thereby maintain radiation losses. This proved unsatisfactory as it was shown qualitatively in Fig. 12, in which it may be seen that indicated temperature fell off, rather than all three thermocouples responding in a single well-defined temperature. Though the same idea of the calorimeter may be used, a new design and refinement is necessary before valid and reproducible results can be obtained by its use.

In order, however, to have some common ground for comparison with results obtained in Part I, the exact temperature curves measured in the present calorimeter which showed very good agreement of the three thermocouples showed the

same variation with exhaust pressure. The variation of exhaust temperature with exhaust pressure for thermocouple No. 3 located nearest the exhaust valve, is shown in Fig. 16. As in Ref. 1, it is seen that there is a linear decrease of exhaust temperature with increasing exhaust pressure at each inlet pressure.

In the last series of runs a General Electric shielded thermocouple was placed in the exhaust stack ahead of the calorimeter. Though the thermocouple located here is subject to the wide theoretical temperature variation mentioned above, it is felt that the actual range of variation is much less due to the heat capacity of the engine cylinder parts, the heavy exhaust stacks used, and the radiation shields of the thermocouple. Here it was found, as may be seen from the curve (T_4) in Fig. 16, that the exhaust temperature appears to be independent of inlet pressure and increases with increasing back pressure. Temperatures in the exhaust stack were found to be several hundred degrees higher than those measured at the calorimeter thermocouples. This trend of increasing temperature with increasing exhaust pressure, as well as the quantitative values of exhaust temperatures, are in fair agreement with values obtained from computation using theoretical thermodynamic charts. (Ref. 10.)

The determination of the energy of exhaust gases to a closer approximation by temperature measurement should be in-

some variation with respect to pressure. The variation of
atmospheric temperature with respect to pressure for atmospheric
air is shown in Fig. 1. The vertical axis is pressure in lb.
per sq. in. absolute, and the horizontal axis is altitude in
feet. It is seen that the temperature of the air increases
with altitude up to about 36,000 feet, and then decreases.
In the case of air a constant density is assumed
throughout the whole range of altitude shown in Fig. 1.
The atmospheric pressure is assumed to be constant at sea
level, and the atmospheric temperature is assumed to be
constant at sea level. It is also assumed that the
atmospheric pressure is constant at sea level, and the
atmospheric temperature is constant at sea level. The
atmospheric pressure is assumed to be constant at sea
level, and the atmospheric temperature is assumed to be
constant at sea level. The atmospheric pressure is
assumed to be constant at sea level, and the atmospheric
temperature is assumed to be constant at sea level.

vestigated further. However, at present, the temperature at any exhaust pressure investigated in this work is still beyond that which any modern turbine can withstand, so the performance of the turbine in a CMT system is not necessarily limited by the exhaust temperature of the internal combustion engine.

COMPRESSOR-ENGINE-TURBINE SYSTEM

While the power output of the engine proper is reduced considerably with increase of the ratio, P_0/P_1 , it is possible to realize a greater overall output by use of a compressor-engine-turbine system. In such a system the engine would be operating at a relatively high P_0/P_1 with considerably reduced volumetric efficiency. However, utilization of a part of the energy available in the exhaust gases through a turbine increases the overall output to such degree that a successful compressor-engine-turbine system appears wholly feasible.

In order to make direct comparison of full power conditions (Ref. 1) and cruising conditions, the total brake horsepower output of a CET system was calculated on the following assumptions:

1. The turbine, engine, and compressor are directly connected to the propeller shaft.
2. The compressor works on the mixture of vaporized fuel and air.
3. Airplane speed is 300 miles per hour, indicated airspeed, and the full effect of ram on pressure and temperature is utilized.
4. Turbine and compressor efficiencies are .70. This appears low for efficiencies of present turbines and compressors but these values are assumed in order to obtain comparison with Ref. 1.

COMPARISON OF THE TWO SYSTEMS

While the power output of the engine system is reduced

considerably with increase of the ratio, the efficiency is

reduced as well as a greater weight output is required of a

compressor-engine system. In such a system the en-

gine would be operated at a relatively high RPM with

considerably reduced volumetric efficiency. However, still-

more of a part of the energy available in the exhaust gases

through a turbine increases the overall output to such extent

that a successful compressor-engine-turbine system appears

wholly feasible.

In order to make direct comparison of full power condi-

tions (Net. 1) and various conditions, the total brake

horsepower output of a GT system was calculated on the follow-

ing assumptions:

1. The turbine, engine, and compressor are directly connected to the propeller shaft.
2. The compressor works on the basis of isentropic compression of inlet air.
3. Airplane speed is 300 miles per hour, indicated airspeed, and the full thrust of ram on pressure and temperature is utilized.
4. Turbine and compressor efficiencies are .70. This appears low for efficiencies of present engines and compressors but these values are assumed in order to obtain comparison with Net. 1.

5. The system is operating in a standard atmosphere.
6. The highest measured temperature in the calorimeter is the temperature of the gas to the turbine, and the turbine is capable of operating at these temperatures. No correction to exhaust temperature is made for variation in inlet temperature from 140°F . The variation in exhaust temperature due to this effect is considered to be within the range of error in measuring the exhaust temperature.
7. The net total power of the CET system is equal to the measured brakehorsepower of the engine minus the computed power absorbed by the compressor plus the computed power delivered by the turbine.

The variation of net total brakehorsepower with engine exhaust pressure is shown in Fig. 17, at sea level, 15,000 feet and 30,000 feet. Cruising power output appears to be about 70% of full power output (Ref. 1). In Fig. 18 the variation of net total brakehorsepower is plotted against the ratio P_0/P_1 . The optimum P_0/P_1 increases with increase in altitude and occurs between the values .75 and 1.0. This is consistent with results obtained in Refs. 1 and 7.

Table II shows the work per pound of air for the various components of the CET system. The power absorbed by the compressor remains constant for any one condition of inlet pressure and altitude, i.e., does not change with variation in exhaust pressure. The turbine work increases with increase

3. The system is operating in a standard atmosphere.

4. The element measured is representative in the entire water in the temperature of the gas in the turbine and the turbine is capable of operating at these temperatures. No correction is required for temperature. There is a note for variation in inlet temperature from 140°F. The variation in element temperature due to this effect is considered to be within the range of error in measuring the element temperature.

5. The net total power of the GT system is equal to the measured shaft horsepower of the engine minus the corrected power absorbed by the compressor plus the corrected power delivered to the turbine.

The variation of net total shaft horsepower with engine exhaust pressure is shown in Fig. 17, at sea level, 15,000 feet and 30,000 feet. Corrected power output appears to be about 70% of full power output (net). In Fig. 18 the variation of net total shaft horsepower is plotted against the ratio P_2/P_1 . The optimum P_2/P_1 increases with increase in altitude and occurs between the values .75 and 1.0. This is consistent with results obtained in Figs. 1 and 7.

Table II shows the work per pound of air for sea values components of the GT system. The power absorbed by the compressor turbine is constant for any one condition of inlet pressure and altitude, i.e., does not change with variation in exhaust pressure. The turbine work increases with increase

in engine exhaust pressure. It also increases with altitude in greater proportion than does the power absorbed by the compressor. At $P_1 = 50''$ Hg., $P_e = 60''$ Hg., the power delivered by the turbine, in excess of the power absorbed by the compressor, accounts for approximately 18% of the net total power. This value would be somewhat reduced in a multiple cylinder arrangement because the percentage of power absorbed by engine friction would be reduced. The multiple cylinder arrangement would result in higher total output.

A more valid evaluation of the experimental results may be realized by analyzing the CET system from the viewpoint of mean effective pressures and specific fuel consumption. Ref. 2 analyzes the full power experimental results from this viewpoint. A better comparison of full power conditions and cruising conditions may be obtained with such an analysis for cruising conditions. Calculations are made for 15,000 feet and 30,000 feet, based on the same assumptions as Ref. 2:

Compressor efficiency:	$\approx .85$
Turbine efficiency:	$\approx .90$
Temperature of gases entering turbine:	From curves in Fig. 16
Atmospheric conditions:	Standard
Ram pressure at compressor intake:	1" Hg.
Engine BMEP corrected from:	Fig. 4
Friction m.e.p.	Estimated, based on Fig. 5, Ref. 2

in engine exhaust pressure. It also increases with air-
 flow in transfer operation from the power absorbed by
 the compressor. At a 100% air flow, the power
 delivered by the turbine, in excess of the power absorbed
 by the compressor, amounts to approximately 10% of the
 net total power. This value would be somewhat reduced in
 a multiple cylinder arrangement because the percentage of
 power absorbed by engine friction would be reduced. The
 multiple cylinder arrangement would result in higher total
 output.

A more valid evaluation of the experimental results
 may be obtained by analyzing the test system from the view-
 point of mean effective pressure and specific fuel con-
 sumption. Ref. 2 analyzes the full power experimental re-
 sults from this viewpoint. A better comparison of full
 power conditions and existing conditions may be obtained
 when such an analysis for existing conditions. Calculations
 are made for 15,000 rpm and 10,000 rpm, based on the same
 assumptions as Ref. 2.

Compressor efficiency:	0.82
Turbine efficiency:	0.90
Temperature of gases entering turbine:	from curves in Fig. 16
Atmospheric conditions:	Standard
Low pressure at compressor inlet:	14.7 psia
Engine with corrected time:	110.4
Friction M.E.P.:	Estimated, based on Fig. 2, Ref. 2

An inlet pressure of 40" Hg. was selected as representative of cruising conditions. Test results were corrected to a compression ratio of 6.5 and an inlet temperature of 586°R. Values for exhaust pressures to 80" Hg. were extrapolated. Methods for calculation are shown under FORMULAE AND SAMPLE CALCULATIONS.

Fig. 19 shows the component mean effective pressures and the net m.e.p. under cruising conditions at 15,000 feet and 30,000 feet. Maximum net m.e.p. occurs at an engine exhaust pressure of about 40" Hg. Turbine m.e.p. increases with altitude while BMEP decreases linearly.

Fig. 20 compares component and net mean effective pressures at full power and cruising conditions at 30,000 feet. Full power mean effective pressures were taken from Ref. 2. It was expected that engine BMEP at cruising conditions would slightly exceed the engine BMEP at full power due to increased volumetric efficiency at cruising conditions. There is considerable divergence in the two curves as exhaust pressure is increased. This may be due to volumetric efficiency characteristics, i.e., volumetric efficiency at high piston speed may be reduced at a greater rate with increasing exhaust pressure than at a lower piston speed. The cruising condition IMEP values were taken directly from values determined from the indicator card. Full power IMEP values were taken from a general curve of IMEP/inlet pressure vs. P_0/P_1 . This could possibly ac-

An inlet pressure of 40" Hg. was selected as representative of existing conditions. Test results were computed to a compression ratio of 6.5 and an inlet temperature of 580° R. Within the engine pressure to 60" Hg. was maintained. Methods for calculating the same were presented.

Calculation.

Fig. 12 shows the engine mean effective pressure and the net m.e.p. under existing conditions at 15,000 rpm and 30,000 rpm. Within the m.e.p. curve at an engine speed of 40" Hg. the net m.e.p. is indicated with a side view of the engine.

Fig. 13 compares engine and net mean effective pressure at full power and existing conditions at 30,000 rpm. Full power mean effective pressure was taken from Fig. 12. It was expected that engine EHP at existing conditions would slightly exceed the engine EHP at full power. The difference volumetric efficiency at existing conditions. There is considerable divergence in the two curves as engine speed is increased. This may be due to volumetric efficiency characteristics, i.e., volumetric efficiency at high speed may be reduced as a greater rate of piston speed is required than at a lower piston speed. The existing conditions EHP values were taken directly from values determined from the indicated card. Full power EHP values were taken from a general curve of EHP/indicated pressure vs. rpm. This would possibly be-

count for the difference in slope of the BMEP curves.

Since the compressor and turbine m.e.p. were essentially the same at full power and cruising, the net m.e.p. varies about the same as engine BMEP, i.e., the net m.e.p. is somewhat higher for cruising conditions.

Net specific fuel consumption at full power and cruising at 30,000 feet are compared in Fig. 21. The net s.f.c. is significantly reduced at cruising.

The results of the temperature measurements in the cylinder exhaust (T_4 , Fig. 16) indicate that energy is available in such amount that it would not be unreasonable to assume an arrangement whereby the exhaust gases are available to the turbine at all times, at temperature above that at which modern turbines can operate. Assume further a thermostatically controlled intercooler so that gases enter the turbine at 1500°F . This figure (1500°F) is based on the General Electric I-40 engine (Ref. 11) with some allowance for improved turbine blading material in the future.

Fig. 22 compares the cruising net m.e.p. of such a system with the system previously assumed at 30,000 feet. Net m.e.p. is increased a small but significant amount with increased exhaust pressure and with increased altitude. The turbine m.e.p. furnishes a larger percentage of the total net m.e.p.

count for the difference in slope of the two curves.
Since the compressor and turbine are essentially
the same as before and operating, the net a.s.p. will be
about the same as before, i.e., the net a.s.p. is about
the same for similar conditions.
The specific fuel consumption at full power and cruising
is 30,000 lbs per hour and is 24.5. The net a.s.p. is
slightly reduced as shown.

The results of the temperature measurements in the cylinder
are shown in Fig. 10. The indicated gas temperature is available in
such amount that it would not be unreasonable to assume an at-
mospheric pressure. The exhaust gases are available to the turbine
at all times, as indicated above that at which modern turbines
can operate. Above rather a characteristically controlled inter-
mediate to high power and the turbine at 1500° F. This figure
(1500° F) is based on the General Electric X-45 engine (Fig. 11)
with some allowance for lagged turbine blade material in
the turbine.

Fig. 12 compares the turbine net a.s.p. of such a system
with the system previously assumed at 30,000 lbs. per hour. The net a.s.p.
is increased a small but significant amount with increased tur-
bine pressure and with increased altitude. The turbine a.s.p.
provides a larger percentage of the total net a.s.p.

CONCLUSIONS

As a result of this investigation of the effects of high exhaust pressures on the performance of an internal combustion engine, the following conclusions may be drawn:

1. In the range investigated, the mass rate of air-flow decreases linearly with increase in exhaust pressure.
2. Volumetric efficiency falls off linearly with increase in exhaust pressure.
3. Brake Horsepower decreases almost linearly with increase in exhaust pressure.
4. BMEP and IMEP decrease linearly with increase in exhaust pressure.
5. Indicated Horsepower increases linearly with mass airflow.
6. Mechanical FMEP and PMEP increase linearly with increase in exhaust pressure.
7. FMEP decreases linearly and mechanical FMEP increases with increase in inlet pressure.
8. Brake specific air consumption increases with increase in exhaust pressure, the rate of increase becoming greater with larger values of exhaust pressure.
9. Exhaust temperature increases linearly with increase in exhaust pressure.

CONCLUSIONS

As a result of this investigation of the effects of high
exhaust pressure on the performance of an internal combustion
engine, the following conclusions may be drawn:

1. In the range investigated, the mean rate of air-
flow increases linearly with increase in exhaust
pressure.
2. Indicated efficiency falls off linearly with in-
crease in exhaust pressure.
3. Indicated horsepower decreases almost linearly with
increase in exhaust pressure.
4. BHP and LHP decrease linearly with increase in
exhaust pressure.
5. Indicated horsepower increases linearly with mean
effective pressure.
6. Mechanical efficiency and LHP increase linearly with
increase in exhaust pressure.
7. LHP decreases linearly and mechanically BHP in-
creases with increase in inlet pressure.
8. Brake specific air consumption increases with
increase in exhaust pressure. The rate of increase
becoming greater with larger values of exhaust pres-
sure.
9. Exhaust temperature increases linearly with in-
crease in exhaust pressure.

10. Optimum engine P_e/P_1 for the CET system increases with altitude and occurs between the values .75 and 1.0
11. Net mean effective pressure of the CET system is slightly higher at cruising conditions than at full power conditions.
12. Net specific fuel consumption for the CET system is lower at cruising conditions than at full power conditions.
13. The turbine output in a CET system is more likely to be limited by the turbine blading material than by the energy available in the exhaust gases.

10. The engine design 94/37 for the 1000 hp engine
with 1000 hp and 1000 hp and 1000 hp and 1000 hp
11. The mean electric power of the 1000 hp engine is
slightly higher at existing conditions than at full power
conditions.

12. The electric fuel consumption for the 1000 hp engine
is lower at existing conditions than at full power con-
ditions.

13. The engine output in a 1000 hp engine is more likely
to be limited by the engine design output than by
the energy available in the exhaust gases.

14. The engine output in a 1000 hp engine is more likely
to be limited by the engine design output than by
the energy available in the exhaust gases.

15. The engine output in a 1000 hp engine is more likely
to be limited by the engine design output than by
the energy available in the exhaust gases.

16. The engine output in a 1000 hp engine is more likely
to be limited by the engine design output than by
the energy available in the exhaust gases.

17. The engine output in a 1000 hp engine is more likely
to be limited by the engine design output than by
the energy available in the exhaust gases.

18. The engine output in a 1000 hp engine is more likely
to be limited by the engine design output than by
the energy available in the exhaust gases.

REFERENCES

1. KENNA, W.E., ANTONIAK, C., McCUTCHEON, K.B., - "THE EFFECT OF HIGH EXHAUST PRESSURES ON ENGINE PERFORMANCE AND THE AVAILABILITY OF ENERGY IN EXHAUST GASES AT HIGH PRESSURES", June 1944.
2. TAYLOR, C.F., Massachusetts Institute of Technology - "EFFECT OF ENGINE EXHAUST PRESSURE ON PERFORMANCE OF COMPRESSOR-ENGINE-TURBINE UNITS", SAE Journal, Vol. 54, No. 2, February 1946.
3. "FLUID METERS, THEIR THEORY AND APPLICATION" Part I, ASME Research Publication, 4th Edition, 1937.
4. "ASME POWER TEST CODES OF 1940" Part 5, Chapter 4, FLOW MEASUREMENT, July 1940.
5. "TEMPERATURE - ITS MEASUREMENT AND CONTROL IN SCIENCE AND INDUSTRY", Reinhold Publishing Corp., 1941.
6. KING, W.J. - "MEASUREMENT OF HIGH TEMPERATURES IN HIGH VELOCITY GAS STREAMS" - Transactions, ASME, vol. 65, 1943, pp. 421 - 431.
7. PINKEL, BENJAMIN, - "PACA STUDY OF THE UTILIZATION OF EXHAUST GAS OF AIRCRAFT ENGINES", SAE Reprint of paper presented at SAE National Aeronautic Meeting, April 1946.
8. ROHSENOW, W.M. - "A GRAPHICAL DETERMINATION OF UNSHIELDED THERMOCOUPLE THERMAL CORRECTION", Transactions, ASME, vol. 68, 1946, pp. 195 - 198.
9. NEAFAMS, W.H. - "HEAT TRANSMISSION" - 2nd Edition, McGraw, Hill Book Company, New York, 1942.

REFERENCES

1. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
2. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
3. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
4. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
5. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
6. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
7. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
8. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
9. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."
10. KERN, W.R., ACTIVITY, D., INVENTION, K.S., - "THE EFFECT OF THE KERN METHOD OF RADIATION THERAPY ON THE EFFICIENCY OF RADIATION IN KERN'S CASE AT THE UNIVERSITY, 1944."

REFERENCES (Continued)

10. TAYLOR, C.F., TAYLOR, H.G. - "THE INTERNAL COMBUSTION ENGINE", International Textbook Company, 1938.
11. STREIF, DALE E. - "JET PROPULSION GAS TURBINE", Automotive and Aviation Industries, Jan. 1, 1946.
12. KEENAN and KAYE - "A TABLE OF THERMODYNAMIC PROPERTIES OF AIR".

FORMULAE AND SAMPLE CALCULATIONS

AIR METER:

$$W_A = AKY \sqrt{2g\rho\Delta P}$$

Orifice characteristics:

Diameter = 1.378 inches

Area = 1.490 square inches

K = 0.617

Y = 1.0

$\beta = 0.45$

$$W_A = 3600 \times 1.490 \times 0.617 \times 1.0 \sqrt{\frac{2 \times 386 \times p \times 70.7}{1728 \times T \times 53.3} \times \frac{H \times 1.00 \times 62.4}{1728}}$$

$$W_A = 485 \sqrt{\frac{P \times H}{T}} \quad \text{lb/hr}$$

1.00 = specific gravity of water

62.4 = weight of water, lb/cu.ft.

70.7 = conversion, "Hg to p.s.f.

53.3 = gas constant

P = pressure in front of orifice, in inches of mercury; equal to gage-pressure in front of orifice plus barometer (both in inches of mercury).

H = orifice pressure drop in inches of water.

T = temperature of air entering orifice in degrees Rankine.

VOLUMETRIC EFFICIENCY:

$$\epsilon = \frac{W_A}{n V \rho_1}$$

$$W_A = \frac{485}{60} \sqrt{\frac{P \times H}{T}} = \text{lbs air/min.}$$

$$n = \frac{2025}{2} = 1012.5 \text{ suction strokes/min.}$$

$$V = \frac{\pi}{4} \times \frac{(5.25)^2 \times 6.25}{1728} = \text{displacement volume, cu.ft.}$$

$$\rho_1 = \text{inlet air density} = \frac{p_1}{T_1} \times \frac{70.7}{53.3}$$

p_1 = pressure in intake manifold, "Hg.

T_1 = inlet air temperature, deg. Rankine.

Then:

$$\epsilon = \frac{W_A/60}{1012.5 \times \frac{\pi}{4} \times \frac{(5.25)^2 \times 6.25}{1728} \times \frac{p_1}{T_1} \times \frac{70.7}{53.3}}$$

$$\epsilon = .0001585 \frac{W_A T_1}{p_1}$$

where W_A = Airflow, lbs/hr.

T_1 = Temperature in intake manifold, deg. Rankine.

p_1 = Manifold pressure, "Hg.

BRAKE HORSEPOWER:

$$\text{Dynamometer arm} = 21.02" = 1.752 \text{ ft.}$$

$$\text{HP} = \frac{2 \pi R N F}{33000}$$

R = Brake arm, ft.

N = rpm

F = Brake load, lbs.

$$\text{Diameter of Hydraulic Piston} = 2.955 \text{ inches}$$

$$\text{Area of Hydraulic Piston} = \frac{\pi}{4} \times (2.955)^2 = 6.85 \text{ sq. in.}$$

HYDRAULIC CALCULATIONS

$$\frac{V}{A} = \frac{1}{2} \sqrt{\frac{2gH}{fL}}$$

$$V = \frac{1}{2} \sqrt{\frac{2gH}{fL}} \times A$$

$$Q = \frac{1}{2} \sqrt{\frac{2gH}{fL}} \times A \times 60$$

$$V = \frac{1}{2} \sqrt{\frac{2gH}{fL}} \times A \times 60 \times 60$$

$$L = \frac{V^2}{2gH} \times \frac{1}{f}$$

Where H = pressure in inches water

f = friction coefficient, see section

Notes:

$$Q = \frac{1}{2} \sqrt{\frac{2gH}{fL}} \times A \times 60 \times 60$$

$$L = \frac{V^2}{2gH} \times \frac{1}{f}$$

where V = velocity, ft/sec

H = pressure in inches water, see section

f = friction coefficient, see section

PIPE SIZING

$$D = \sqrt{\frac{4Q}{\pi V}}$$

$$V = \frac{Q}{A}$$

$$A = \frac{\pi D^2}{4}$$

$$V = \frac{Q}{\frac{\pi D^2}{4}}$$

Where D = diameter of pipe in inches

$$Q = \frac{\pi D^2 V}{4}$$

Force on Piston,

$$F = p \times a$$

p = psi

a = area, inches

$$F = 6.85 p$$

$$1 \text{ psi} = 2.042 \text{ "Hg}$$

$$F = \frac{h}{2.042} \times 6.85 = 3.35 h$$

h = "Hg

$$N = 2025$$

Then

$$Bhp = \frac{2\pi \times 1.752 \times 2025 \times 3.35 h}{33000} \quad h = \text{brake load, "Hg}$$

$$Bhp = 2.26 h$$

EMEP:

$$HP = \frac{PLAN}{33000} = \frac{2\pi ENP}{33000}$$

P = EMEP, psia.

L = stroke, ft.

A = piston area, sq. in.

$$P = \frac{2(2\pi ENP)}{LAN} = \frac{4\pi NP}{LA}$$

$$= \frac{4\pi \times 1.752 \times 12 \times 4 \times 3.35 h}{6.25 \times \pi \times (5.25)^2}$$

$$= 6.55 h$$

h = brake load, "Hg

$\log a = 0.1761$ $\log b = 0.1761$ $\log c = 0.1761$
 $\log d = 0.1761$ $\log e = 0.1761$ $\log f = 0.1761$

$$1 \text{ bel} = 1.000 \text{ bel}$$

$$10 \text{ bel} = 10.000 \text{ bel}$$

$$100 \text{ bel} = 100.000 \text{ bel}$$

$$1000 \text{ bel} = 1000.000 \text{ bel}$$

$$10000 \text{ bel} = 10000.000 \text{ bel}$$

$$\log 10 = 1.0000$$

$$\log 100 = 2.0000$$

$$\log 1000 = 3.0000$$

$$\log 10000 = 4.0000$$

$$\log 100000 = 5.0000$$

$$\log 1000000 = 6.0000$$

$$\log 10000000 = 7.0000$$

SAMPLE CALCULATION OF COT SYSTEM

From Ref. (12).

Subscript 1 = before compressor or turbine

Subscript 2 = after compressor or turbine

h_a = Enthalpy of air, Btu/lb. air

h_f = Enthalpy of fuel, Btu/lb. fuel

h_m = Enthalpy of mixture Btu/lb. mixture

h_{ma} = Enthalpy of mixture per pound of air, in Btu/lb.

h_e = Enthalpy of exhaust gases, Btu/lb. gases

f = Fuel-air ratio = .065

η_c = Compressor efficiency = .70

η_t = Turbine efficiency = .70

$h_a + f h_f = h_{ma1}$

$h_{m1} = \frac{h_{ma1}}{1+f}$ = Enthalpy of mixture entering compressor

Compressor Work = $\frac{h_{m1} - h_{m2}}{.7}$ Btu/lb. mixture

= $\frac{1.08 h_{m2} - h_{m1}}{.7}$ Btu/lb. air

where h_{m2} is obtained from Ref. (12) by means of relative pressure ratio of manifold pressure to atmospheric pressure plus ram.

Engine Work = $\frac{BHP \times 2545}{\text{lbs air/hr}}$ = Btu/lb. air

ANALYSIS OF THE DATA

From Set (12).

1	Efficiency of engine at 1000 r.p.m.	0.25
2	Efficiency of engine at 1500 r.p.m.	0.28
3	Efficiency of engine at 2000 r.p.m.	0.30
4	Efficiency of engine at 2500 r.p.m.	0.32
5	Efficiency of engine at 3000 r.p.m.	0.34
6	Efficiency of engine at 3500 r.p.m.	0.36
7	Efficiency of engine at 4000 r.p.m.	0.38
8	Efficiency of engine at 4500 r.p.m.	0.40
9	Efficiency of engine at 5000 r.p.m.	0.42
10	Efficiency of engine at 5500 r.p.m.	0.44
11	Efficiency of engine at 6000 r.p.m.	0.46
12	Efficiency of engine at 6500 r.p.m.	0.48
13	Efficiency of engine at 7000 r.p.m.	0.50
14	Efficiency of engine at 7500 r.p.m.	0.52
15	Efficiency of engine at 8000 r.p.m.	0.54
16	Efficiency of engine at 8500 r.p.m.	0.56
17	Efficiency of engine at 9000 r.p.m.	0.58
18	Efficiency of engine at 9500 r.p.m.	0.60
19	Efficiency of engine at 10000 r.p.m.	0.62
20	Efficiency of engine at 10500 r.p.m.	0.64
21	Efficiency of engine at 11000 r.p.m.	0.66
22	Efficiency of engine at 11500 r.p.m.	0.68
23	Efficiency of engine at 12000 r.p.m.	0.70
24	Efficiency of engine at 12500 r.p.m.	0.72
25	Efficiency of engine at 13000 r.p.m.	0.74
26	Efficiency of engine at 13500 r.p.m.	0.76
27	Efficiency of engine at 14000 r.p.m.	0.78
28	Efficiency of engine at 14500 r.p.m.	0.80
29	Efficiency of engine at 15000 r.p.m.	0.82
30	Efficiency of engine at 15500 r.p.m.	0.84
31	Efficiency of engine at 16000 r.p.m.	0.86
32	Efficiency of engine at 16500 r.p.m.	0.88
33	Efficiency of engine at 17000 r.p.m.	0.90
34	Efficiency of engine at 17500 r.p.m.	0.92
35	Efficiency of engine at 18000 r.p.m.	0.94
36	Efficiency of engine at 18500 r.p.m.	0.96
37	Efficiency of engine at 19000 r.p.m.	0.98
38	Efficiency of engine at 19500 r.p.m.	1.00
39	Efficiency of engine at 20000 r.p.m.	1.02
40	Efficiency of engine at 20500 r.p.m.	1.04
41	Efficiency of engine at 21000 r.p.m.	1.06
42	Efficiency of engine at 21500 r.p.m.	1.08
43	Efficiency of engine at 22000 r.p.m.	1.10
44	Efficiency of engine at 22500 r.p.m.	1.12
45	Efficiency of engine at 23000 r.p.m.	1.14
46	Efficiency of engine at 23500 r.p.m.	1.16
47	Efficiency of engine at 24000 r.p.m.	1.18
48	Efficiency of engine at 24500 r.p.m.	1.20
49	Efficiency of engine at 25000 r.p.m.	1.22
50	Efficiency of engine at 25500 r.p.m.	1.24
51	Efficiency of engine at 26000 r.p.m.	1.26
52	Efficiency of engine at 26500 r.p.m.	1.28
53	Efficiency of engine at 27000 r.p.m.	1.30
54	Efficiency of engine at 27500 r.p.m.	1.32
55	Efficiency of engine at 28000 r.p.m.	1.34
56	Efficiency of engine at 28500 r.p.m.	1.36
57	Efficiency of engine at 29000 r.p.m.	1.38
58	Efficiency of engine at 29500 r.p.m.	1.40
59	Efficiency of engine at 30000 r.p.m.	1.42
60	Efficiency of engine at 30500 r.p.m.	1.44
61	Efficiency of engine at 31000 r.p.m.	1.46
62	Efficiency of engine at 31500 r.p.m.	1.48
63	Efficiency of engine at 32000 r.p.m.	1.50
64	Efficiency of engine at 32500 r.p.m.	1.52
65	Efficiency of engine at 33000 r.p.m.	1.54
66	Efficiency of engine at 33500 r.p.m.	1.56
67	Efficiency of engine at 34000 r.p.m.	1.58
68	Efficiency of engine at 34500 r.p.m.	1.60
69	Efficiency of engine at 35000 r.p.m.	1.62
70	Efficiency of engine at 35500 r.p.m.	1.64
71	Efficiency of engine at 36000 r.p.m.	1.66
72	Efficiency of engine at 36500 r.p.m.	1.68
73	Efficiency of engine at 37000 r.p.m.	1.70
74	Efficiency of engine at 37500 r.p.m.	1.72
75	Efficiency of engine at 38000 r.p.m.	1.74
76	Efficiency of engine at 38500 r.p.m.	1.76
77	Efficiency of engine at 39000 r.p.m.	1.78
78	Efficiency of engine at 39500 r.p.m.	1.80
79	Efficiency of engine at 40000 r.p.m.	1.82
80	Efficiency of engine at 40500 r.p.m.	1.84
81	Efficiency of engine at 41000 r.p.m.	1.86
82	Efficiency of engine at 41500 r.p.m.	1.88
83	Efficiency of engine at 42000 r.p.m.	1.90
84	Efficiency of engine at 42500 r.p.m.	1.92
85	Efficiency of engine at 43000 r.p.m.	1.94
86	Efficiency of engine at 43500 r.p.m.	1.96
87	Efficiency of engine at 44000 r.p.m.	1.98
88	Efficiency of engine at 44500 r.p.m.	2.00
89	Efficiency of engine at 45000 r.p.m.	2.02
90	Efficiency of engine at 45500 r.p.m.	2.04
91	Efficiency of engine at 46000 r.p.m.	2.06
92	Efficiency of engine at 46500 r.p.m.	2.08
93	Efficiency of engine at 47000 r.p.m.	2.10
94	Efficiency of engine at 47500 r.p.m.	2.12
95	Efficiency of engine at 48000 r.p.m.	2.14
96	Efficiency of engine at 48500 r.p.m.	2.16
97	Efficiency of engine at 49000 r.p.m.	2.18
98	Efficiency of engine at 49500 r.p.m.	2.20
99	Efficiency of engine at 50000 r.p.m.	2.22
100	Efficiency of engine at 50500 r.p.m.	2.24

$$\text{Turbine Work} = .7(h_{e1} - h_{e2}) \text{ Btu/lb. exhaust gases}$$

$$= 1.080 \times .7(h_{e1} - h_{e2}) \text{ Btu/lb. air}$$

where h_{e1} is obtained from exhaust

temperature and h_{e2} by relative pressure ratio

of exhaust pressure to atmospheric pressure.

$$\text{Net System Work / lb. air} = \text{Engine Work} - \text{Compressor Work} + \text{Turbine Work}$$

$$\text{Net Horsepower} = \frac{\text{Net Work / lb. air} \times \text{lbs. air/hr.}}{2545}$$

Example:

At: 300 m.p.h., indicated air speed

15000 feet altitude

50" Hg. inlet pressure

40" Hg. exhaust pressure

$$P_a = 16.88 + \frac{.002378}{70.7} \times (300 \times 1.47)^2 = 18.94 \text{ "Hg.}$$

$$T_a = 465 + \frac{(300 \times 1.47)^2}{12,000} = 481^\circ \text{Fabs.}$$

$$h_a = 19.42$$

$$h_f = .5T_f - 375 = .5 \times 481 - 375 = -134.5$$

$$h_{m1} = 19.42 - \frac{(.065 \times 134.5)}{1.065} = \frac{10.69}{1.065} = 10.05$$

	p	h_m	pr
(1)	18.94	10.05	1.417
(2)	50	<u>43.93</u>	3.74
$h_{m2} - h_{m1}$	=	33.88	

$\frac{1}{1.000} \times 1.000 = 1.000$ (1.000 - 1.000) = 0.000

$\frac{1}{1.000} \times 1.000 = 1.000$ (1.000 - 1.000) = 0.000

... to obtain ...

... by relative pressure ratio

of constant pressure in atmospheric pressure.

For system with 1.000 atm = 1.000 atm - 1.000 atm

1.000 atm

1.000 atm = 1.000 atm - 1.000 atm

Results:

1.000 atm, indicated air pressure

1.000 atm, indicated air pressure

1.000 atm, indicated air pressure

1.000 atm, indicated air pressure

$$P_2 = 1.000 + \frac{1}{1.000} \times (1.000 - 1.000) = 1.000 \text{ atm}$$

$$P_4 = 1.000 + \frac{1}{1.000} \times (1.000 - 1.000) = 1.000 \text{ atm}$$

$$P_4 = 1.000$$

$$P_2 = 1.000 - 1.000 = 0.000$$

$$P_2 = 1.000 - 1.000 = 0.000$$

1.000	1.000	1.000	(1)
1.000	1.000	1.000	(2)
1.000	1.000	1.000	

$$\text{Compressor Work} = 1.065 \times \frac{33.88}{.7} = 51.4 \text{ Btu/lb. air.}$$

$$\text{Engine Work} = \frac{72 \times 2545}{496} = 369 \text{ Btu/lb. air}$$

Turbine Work:

Exhaust Temperature: 2024°Fabs.

	T	P	h_e	Pr
(1)	2024	40" Hg.	416.2	378.3
(2)		29.92	<u>308.4</u>	159.5

$$h_{e1} - h_{e2} = 107.8$$

$$\text{Turbine Work} = 1.065 \times .7 \times 107.8 = 80.2$$

$$\text{Net Work} = 369 - 51.4 + 80.2 = 398 \text{ Btu/lb. air}$$

$$\text{Net Power} = \frac{398 \times 496}{2545} = 78.1 \text{ Hp } \#$$

Temperature = $1.00 \times 10^3 \times 1.1 \times 10^{-3} = 1.1 \times 10^0 = 1.1$

Pressure = $1.1 \times 10^3 \times 1.1 \times 10^{-3} = 1.1 \times 10^0 = 1.1$

Volume

Exhaust Temperature = $1.1 \times 10^3 \times 1.1 \times 10^{-3} = 1.1 \times 10^0 = 1.1$

	1	2	3	4
(1)	1.1	1.1	1.1	1.1
(2)	1.1	1.1	1.1	1.1

Temperature = $1.00 \times 10^3 \times 1.1 \times 10^{-3} = 1.1 \times 10^0 = 1.1$

Pressure = $1.1 \times 10^3 \times 1.1 \times 10^{-3} = 1.1 \times 10^0 = 1.1$

Volume = $1.1 \times 10^3 \times 1.1 \times 10^{-3} = 1.1 \times 10^0 = 1.1$

CALCULATION OF M.E.P. AND S.F.C. FOR CET SYSTEM

M.E.P.:

$$\text{Net m.e.p.} = \text{IMEP} - \text{FMEP} - \text{CMEP} + \text{TMEP}$$

where IMEP is corrected from Fig. 4.

FMEP is estimated, based on Fig. 5, Ref. 2.

$$\text{CMEP} = \frac{2 \times \text{compressor work per unit time}}{\text{engine displacement per unit time}}$$

$$\text{TMEP} = \frac{2 \times \text{turbine work per unit time}}{\text{engine displacement per unit time}}$$

Corrected IMEP = test IMEP [(Fig. 4) from area of indicator diagram, not including pumping cycle] $\times \sqrt{\frac{600}{586} \times \frac{.526}{.513}}$ where .526 is air cycle efficiency at 6.0 compression ratio. The complete correction factor is 1.04.

No correction made on exhaust gas temperature due to increase in compression ratio.

COMPRESSOR INLET CONDITIONS (1" Hg. ram)

Altitude	P _b , "Hg. abs.	T _b , °R
15,000	17.9	472
30,000	9.9	424

Engine Inlet: P_i = 40" Hg. abs.; Inlet manifold temperature, 586°R

Engine Exhaust Pressure: 30" - 80" Hg. abs.

COMPRESSOR M.E.P.:

$$\text{CMEP} = \frac{778}{144} P_i \cdot c_p T_b \left(\frac{P_i}{P_b \times .98} \right)^{.283} - 1 \frac{1}{\eta_c}$$

where: c_p = specific heat of air at constant pressure
= .24 Btu/lb-°F

1900-1901

1990

1957-1958 - 1959 - 1960 - 1961 - 1962 - 1963 - 1964 - 1965 - 1966 - 1967 - 1968 - 1969 - 1970 - 1971 - 1972 - 1973 - 1974 - 1975 - 1976 - 1977 - 1978 - 1979 - 1980 - 1981 - 1982 - 1983 - 1984 - 1985 - 1986 - 1987 - 1988 - 1989 - 1990 - 1991 - 1992 - 1993 - 1994 - 1995 - 1996 - 1997 - 1998 - 1999 - 2000 - 2001 - 2002 - 2003 - 2004 - 2005 - 2006 - 2007 - 2008 - 2009 - 2010 - 2011 - 2012 - 2013 - 2014 - 2015 - 2016 - 2017 - 2018 - 2019 - 2020 - 2021 - 2022 - 2023 - 2024 - 2025 - 2026 - 2027 - 2028 - 2029 - 2030 - 2031 - 2032 - 2033 - 2034 - 2035 - 2036 - 2037 - 2038 - 2039 - 2040 - 2041 - 2042 - 2043 - 2044 - 2045 - 2046 - 2047 - 2048 - 2049 - 2050 - 2051 - 2052 - 2053 - 2054 - 2055 - 2056 - 2057 - 2058 - 2059 - 2060 - 2061 - 2062 - 2063 - 2064 - 2065 - 2066 - 2067 - 2068 - 2069 - 2070 - 2071 - 2072 - 2073 - 2074 - 2075 - 2076 - 2077 - 2078 - 2079 - 2080 - 2081 - 2082 - 2083 - 2084 - 2085 - 2086 - 2087 - 2088 - 2089 - 2090 - 2091 - 2092 - 2093 - 2094 - 2095 - 2096 - 2097 - 2098 - 2099 - 2100 - 2101 - 2102 - 2103 - 2104 - 2105 - 2106 - 2107 - 2108 - 2109 - 2110 - 2111 - 2112 - 2113 - 2114 - 2115 - 2116 - 2117 - 2118 - 2119 - 2120 - 2121 - 2122 - 2123 - 2124 - 2125 - 2126 - 2127 - 2128 - 2129 - 2130 - 2131 - 2132 - 2133 - 2134 - 2135 - 2136 - 2137 - 2138 - 2139 - 2140 - 2141 - 2142 - 2143 - 2144 - 2145 - 2146 - 2147 - 2148 - 2149 - 2150 - 2151 - 2152 - 2153 - 2154 - 2155 - 2156 - 2157 - 2158 - 2159 - 2160 - 2161 - 2162 - 2163 - 2164 - 2165 - 2166 - 2167 - 2168 - 2169 - 2170 - 2171 - 2172 - 2173 - 2174 - 2175 - 2176 - 2177 - 2178 - 2179 - 2180 - 2181 - 2182 - 2183 - 2184 - 2185 - 2186 - 2187 - 2188 - 2189 - 2190 - 2191 - 2192 - 2193 - 2194 - 2195 - 2196 - 2197 - 2198 - 2199 - 2200 - 2201 - 2202 - 2203 - 2204 - 2205 - 2206 - 2207 - 2208 - 2209 - 2210 - 2211 - 2212 - 2213 - 2214 - 2215 - 2216 - 2217 - 2218 - 2219 - 2220 - 2221 - 2222 - 2223 - 2224 - 2225 - 2226 - 2227 - 2228 - 2229 - 2230 - 2231 - 2232 - 2233 - 2234 - 2235 - 2236 - 2237 - 2238 - 2239 - 2240 - 2241 - 2242 - 2243 - 2244 - 2245 - 2246 - 2247 - 2248 - 2249 - 2250 - 2251 - 2252 - 2253 - 2254 - 2255 - 2256 - 2257 - 2258 - 2259 - 2260 - 2261 - 2262 - 2263 - 2264 - 2265 - 2266 - 2267 - 2268 - 2269 - 2270 - 2271 - 2272 - 2273 - 2274 - 2275 - 2276 - 2277 - 2278 - 2279 - 2280 - 2281 - 2282 - 2283 - 2284 - 2285 - 2286 - 2287 - 2288 - 2289 - 2290 - 2291 - 2292 - 2293 - 2294 - 2295 - 2296 - 2297 - 2298 - 2299 - 2300 - 2301 - 2302 - 2303 - 2304 - 2305 - 2306 - 2307 - 2308 - 2309 - 2310 - 2311 - 2312 - 2313 - 2314 - 2315 - 2316 - 2317 - 2318 - 2319 - 2320 - 2321 - 2322 - 2323 - 2324 - 2325 - 2326 - 2327 - 2328 - 2329 - 2330 - 2331 - 2332 - 2333 - 2334 - 2335 - 2336 - 2337 - 2338 - 2339 - 2340 - 2341 - 2342 - 2343 - 2344 - 2345 - 2346 - 2347 - 2348 - 2349 - 2350 - 2351 - 2352 - 2353 - 2354 - 2355 - 2356 - 2357 - 2358 - 2359 - 2360 - 2361 - 2362 - 2363 - 2364 - 2365 - 2366 - 2367 - 2368 - 2369 - 2370 - 2371 - 2372 - 2373 - 2374 - 2375 - 2376 - 2377 - 2378 - 2379 - 2380 - 2381 - 2382 - 2383 - 2384 - 2385 - 2386 - 2387 - 2388 - 2389 - 2390 - 2391 - 2392 - 2393 - 2394 - 2395 - 2396 - 2397 - 2398 - 2399 - 2400 - 2401 - 2402 - 2403 - 2404 - 2405 - 2406 - 2407 - 2408 - 2409 - 2410 - 2411 - 2412 - 2413 - 2414 - 2415 - 2416 - 2417 - 2418 - 2419 - 2420 - 2421 - 2422 - 2423 - 2424 - 2425 - 2426 - 2427 - 2428 - 2429 - 2430 - 2431 - 2432 - 2433 - 2434 - 2435 - 2436 - 2437 - 2438 - 2439 - 2440 - 2441 - 2442 - 2443 - 2444 - 2445 - 2446 - 2447 - 2448 - 2449 - 2450 - 2451 - 2452 - 2453 - 2454 - 2455 - 2456 - 2457 - 2458 - 2459 - 2460 - 2461 - 2462 - 2463 - 2464 - 2465 - 2466 - 2467 - 2468 - 2469 - 2470 - 2471 - 2472 - 2473 - 2474 - 2475 - 2476 - 2477 - 2478 - 2479 - 2480 - 2481 - 2482 - 2483 - 2484 - 2485 - 2486 - 2487 - 2488 - 2489 - 2490 - 2491 - 2492 - 2493 - 2494 - 2495 - 2496 - 2497 - 2498 - 2499 - 2500 - 2501 - 2502 - 2503 - 2504 - 2505 - 2506 - 2507 - 2508 - 2509 - 2510 - 2511 - 2512 - 2513 - 2514 - 2515 - 2516 - 2517 - 2518 - 2519 - 2520 - 2521 - 2522 - 2523 - 2524 - 2525 - 2526 - 2527 - 2528 - 2529 - 2530 - 2531 - 2532 - 2533 - 2534 - 2535 - 2536 - 2537 - 2538 - 2539 - 2540 - 2541 - 2542 - 2543 - 2544 - 2545 - 2546 - 2547 - 2548 - 2549 - 2550 - 2551 - 2552 - 2553 - 2554 - 2555 - 2556 - 2557 - 2558 - 2559 - 2560 - 2561 - 2562 - 2563 - 2564 - 2565 - 2566 - 2567 - 2568 - 2569 - 2570 - 2571 - 2572 - 2573 - 2574 - 2575 - 2576 - 2577 - 2578 - 2579 - 2580 - 2581 - 2582 - 2583 - 2584 - 2585 - 2586 - 2587 - 2588 - 2589 - 2590 - 2591 - 2592 - 2593 - 2594 - 2595 - 2596 - 2597 - 2598 - 2599 - 2600 - 2601 - 2602 - 2603 - 2604 - 2605 - 2606 - 2607 - 2608 - 2609 - 2610 - 2611 - 2612 - 2613 - 2614 - 2615 - 2616 - 2617 - 2618 - 2619 - 2620 - 2621 - 2622 - 2623 - 2624 - 2625 - 2626 - 2627 - 2628 - 2629 - 2630 - 2631 - 2632 - 2633 - 2634 - 2635 - 2636 - 2637 - 2638 - 263

where \mathbf{C}_{eff} is defined as

W. 100, 2, 317 on hand, Volume at 100

[illegible]

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$$\text{side } \frac{45}{12} = \frac{15}{4} \Rightarrow \left[\text{also using} \right]$$

the O_2 to generate the slip at x_0 .

gives rise to the following

• 10 •

It is not possible to make an exact comparison of the two methods.

1997-1998

Year	1950	1951	1952
1950	100.00	100.00	100.00
1951	100.00	100.00	100.00
1952	100.00	100.00	100.00

Engine class: V-8, 400 cc.;
Weight, 200 lb.

Latitude: 50° - 55° N.

$$\frac{1}{2} \times 1 = 0.5, \quad \frac{1}{2} \times \frac{1}{2} = 0.25, \quad 4 \times 0.25 = 1, \quad \frac{1}{2} \times \frac{1}{2} = 0.25, \quad 4 \times 0.25 = 1$$

Indikator 1a bis 1c sind öffentlich & 1d ist intern

$\gamma^0 = \alpha_1 \mu_{\text{max}} / K$

ρ_1 = inlet density ($\rho_1 = P_1/T_1 \times 70.7/53.3$)

e = volumetric efficiency (from Fig. 2)

η_c = compressor efficiency.

The factor .98 is inserted to allow for 2% pressure loss between compressor and engine manifold.

TURBINE M.E.P.:

$$\text{TMEP} = \frac{778}{144} \rho_1 e (h_e - h_o)(1 + r) \eta_t$$

where: h_e = enthalpy at engine exhaust pressure, P_e ,

and temperature T_e , Btu/lb. air

h_o = enthalpy, Btu/lb. air, after reversible

adiabatic expansion from ex-

haust conditions, P_e and T_e ,

to atmospheric pressure.

r = fuel-air ratio.

η_t = Turbine efficiency.

NET S.F.C.:

The indicated specific fuel consumption of the engine was constant at .365 lb. fuel/ihp-hr. The net S.F.C. for the CET system was calculated as follows:

$$\text{Net S.F.C.} = .365 \times \text{IMEP}/\text{net m.e.p.}$$

SAMPLE CALCULATION:

$P_1 = 40''\text{Hg.}; P_e = 40''\text{Hg.}; \text{altitude} = 15,000 \text{ feet};$

$e = .85; \quad \eta_t = .90$

$\text{IMEP} = 194.1 \times 1.04 = 202 \text{ p.s.i.}$

$\text{PMEP} = 26 \text{ p.s.i., (estimated)}$

$\rho_1 = 1.33 \times 40/586 = .0907 \text{ lbs/ft}^3.$

12.22/5.00/21 = 24/21 = 8/7

2 = 2/1 = 2/1

7 = 7/1 = 7/1

The above is in relation to the 12/22/5.00/21

and the 2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

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2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

The indicated number 2/1 is the number of the engine

and the 2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

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2/1 = 2/1

2/1 = 2/1

2/1 = 2/1

$$\phi \text{ (from Fig. 2) } = .884$$

$$\text{CMEP} = \frac{778}{144} (.0907) \times .884 \times .24 \times 472 \left[\left(\frac{40}{17.9 \times .98} \right)^{.283} - 1 \right] \frac{1}{85}$$

$$= 19.6 \text{ p.s.i.}$$

$$T_c \text{ (from Fig. 16) } = 1971^\circ \text{R}$$

$$\text{FMEP} = \frac{778}{144} (.0907) \times .884 \times (401.5 - 296.6) \times .90 \times 1.065$$

$$= 43.4 \text{ p.s.i.}$$

$$\text{Net m.e.p.} = 202 - 26.0 - 19.6 + 43.4 = 199 \text{ p.s.i.}$$

$$\text{Engine BMEP} = \text{IMEP} - \text{FMEP} = 202 - 26.0 = 176.0 \text{ p.s.i.}$$

$$\text{Net s.f.e.} = .365 \times \frac{202}{199} = .371.$$

1. The first part of the problem is to find the value of the function $f(x)$ at the point $x = 1$.

$$f(1) = \frac{1}{2} \left[1 - \frac{1}{2} \right] = \frac{1}{4}$$

2. The second part of the problem is to find the value of the function $f(x)$ at the point $x = 2$.

$$f(2) = \frac{1}{2} \left[1 - \frac{1}{4} \right] = \frac{3}{8}$$

$$f(3) = \frac{1}{2} \left[1 - \frac{1}{8} \right] = \frac{7}{16}$$

3. The third part of the problem is to find the value of the function $f(x)$ at the point $x = 4$.

$$f(4) = \frac{1}{2} \left[1 - \frac{1}{16} \right] = \frac{15}{32}$$

$$f(5) = \frac{1}{2} \left[1 - \frac{1}{64} \right] = \frac{31}{64}$$

$$f(6) = \frac{1}{2} \left[1 - \frac{1}{256} \right] = \frac{255}{512}$$

4. The fourth part of the problem is to find the value of the function $f(x)$ at the point $x = 7$.

$$f(7) = \frac{1}{2} \left[1 - \frac{1}{4096} \right] = \frac{4095}{8192}$$

$$f(8) = \frac{1}{2} \left[1 - \frac{1}{16384} \right] = \frac{16383}{32768}$$

5. The fifth part of the problem is to find the value of the function $f(x)$ at the point $x = 9$.

$$f(9) = \frac{1}{2} \left[1 - \frac{1}{65536} \right] = \frac{65535}{131072}$$

6. The sixth part of the problem is to find the value of the function $f(x)$ at the point $x = 10$.

$$f(10) = \frac{1}{2} \left[1 - \frac{1}{262144} \right] = \frac{262143}{524288}$$

$$f(11) = \frac{1}{2} \left[1 - \frac{1}{1048576} \right] = \frac{1048575}{2097152}$$

$$f(12) = \frac{1}{2} \left[1 - \frac{1}{4194304} \right] = \frac{4194303}{8388608}$$

$$f(13) = \frac{1}{2} \left[1 - \frac{1}{16777216} \right] = \frac{16777215}{33554432}$$

7. The seventh part of the problem is to find the value of the function $f(x)$ at the point $x = 14$.

$$f(14) = \frac{1}{2} \left[1 - \frac{1}{67108864} \right] = \frac{67108863}{134217728}$$

$$f(15) = \frac{1}{2} \left[1 - \frac{1}{268435456} \right] = \frac{268435455}{536870912}$$

$$f(16) = \frac{1}{2} \left[1 - \frac{1}{1073743360} \right] = \frac{1073743359}{2147486720}$$

$$f(17) = \frac{1}{2} \left[1 - \frac{1}{4295034880} \right] = \frac{4295034879}{8590069760}$$

$$f(18) = \frac{1}{2} \left[1 - \frac{1}{17180139520} \right] = \frac{17180139519}{34360279040}$$

ENGINE B.S.R.

BORE 5.25" STROKE 6.25"

COMPRESSION RATIO 6.0

Piston Speed 2100 fpm.

F/A : .065 ± .001

T_i : 140°F ± 1°F

Inlet Oil Press. : 58-70 psi

2025

R.P.M

Inlet Oil Temp. : 150°F

Cylinder Temp. : 175-185°F

SUMMARY OF RECORD RUNS

REMARKS	DATE	TIME	Bar. Corr.	P _i	P _i Bar.	H	T	Roto.	T _i	H _i	h _i	W _e	W _a	F/A	B.L.	BHP	BMEP	e	T ₃	T ₂	T ₁	T ₄
			" Hg	" Hg	" Hg	" Hg	" R	cm/s	" R	" Hg	" Hg	lbs/hr	lbs/hr	-	%	HP	psia		°F	°F	°F	°F
P _i = 30 Re = 30	1/4/6	1545	1 30.00	40	30.40	5.85	542	7.30	600	0	.50	18.0	28.0	.043	16.7	37.7	104.4	.887	31.8	32.8	34.8	43.4
"	4/2	1445	1 29.70	40	30.10	6.20	547	7.65	601	0	40	18.5	28.4	.062	16.6	37.5	108.6	.109	1907	1363	1320	1920
P _i = 30 Re = 40	4/4	1325	2 30.00	35	30.35	5.90	543	6.70	600	0	10.10	17.15	26.7	.043	14.9	33.7	97.7	.847	30.8	24.5	28.3	43.6
"	4/5	1405	1 29.70	35	30.05	5.40	534	6.90	601	0	10.10	17.45	26.6	.0650	15.2	33.4	99.4	.850	1363	1308	1250	1930
P _i = 30 Re = 50	3/26	1715	3 30.00	50	30.50	4.80	544	6.20	600	0	19.10	6.90	25.7	.0650	14.1	31.8	92.3	.748	30.1	24.1	28.0	45.0
"	4/4	1345	3 30.00	35	30.35	4.90	541	6.30	600	0	19.90	6.50	25.4	.0648	14.0	31.6	91.8	.805	1333	1290	1243	2007
P _i = 30 Re = 60	3/26	1745	4 30.00	40	30.40	4.60	544	6.15	601	.05	30.80	6.30	24.9	.0659	11.2	24.3	73.5	.744	28.5	27.3	26.3	-
"	4/5	1530	5 29.70	30	30.00	4.70	540	6.00	600	0	30.20	6.10	24.8	.0650	10.8	24.4	70.6	.741	1265	1214	1171	-
P _i = 40 Re = 30	4/4	1435	5 30.00	10.50	40.50	8.65	535	12.50	600	10.0	.85	25.30	39.3	.0644	25.6	57.8	167.8	.930	34.6	33.8	32.9	44.0
"	4/5	1400	4 29.70	10.55	40.25	8.70	534	12.60	600	10.0	.90	25.50	39.2	.0650	25.4	57.4	166.1	.937	1529	1494	1454	1960
P _i = 40 Re = 40	3/27	1670	3 29.80	10.40	40.20	7.85	541	11.80	599	9.95	9.85	24.35	37.1	.0656	23.5	53.2	158.2	.882	34.2	33.3	32.3	45.0
"	4/4	1520	6 30.00	10.50	40.50	8.05	537	11.70	600	10.0	10.10	24.30	37.7	.0645	23.8	53.8	156.0	.876	1511	1472	1439	2007
P _i = 40 Re = 50	3/27	1645	4 29.80	10.40	40.20	7.20	541	11.00	600	10.0	19.90	23.30	35.5	.0650	22.0	49.7	144.1	.846	33.4	32.4	31.5	45.4
"	4/4	1545	7 30.00	10.40	40.40	7.45	538	11.10	600	10.0	20.00	23.40	36.3	.0645	22.3	50.4	146.0	.848	1476	1433	1393	2007
P _i = 40 Re = 60	4/4	1605	8 30.00	10.45	40.45	6.90	538	10.60	600	10.0	30.10	22.70	34.8	.0652	20.4	46.1	133.8	.828	33.0	32.1	31.1	46.1
"	4/5	1655	5 29.70	10.45	40.15	6.90	534	10.50	599	10.0	30.10	22.6	34.9	.0648	20.1	45.5	131.5	.832	1454	1420	1376	2058
P _i = 50 Re = 30	4/1	1450	1 29.70	20.7	50.70	11.05	535	15.20	601	20.0	1.10	32.0	44.4	.0648	33.2	75.0	217.5	.947	35.4	34.5	34.0	45.0
"	4/4	1635	9 30.00	20.75	50.75	11.35	534	15.90	599	20.0	1.40	32.8	50.4	.0652	33.0	74.7	217.0	.950	1504	1531	1503	2007
P _i = 50 Re = 40	4/1	1520	2 29.10	20.65	50.35	11.15	534	18.10	600	20.0	10.00	31.85	44.7	.0643	32.0	72.2	209.5	.950	35.4	34.8	34.0	45.0
"	4/4	1655	10 30.00	20.65	50.65	11.00	535	18.10	599	20.0	9.90	31.90	44.6	.0644	31.5	71.2	206.5	.942	1504	1538	1503	2007
P _i = 50 Re = 50	4/1	1545	3 29.70	20.60	50.30	10.50	534	17.00	599	20.0	18.80	30.80	48.2	.0642	30.1	67.4	197.0	.917	34.8	34.0	33.3	45.4
"	4/4	1715	11 30.00	20.60	50.60	10.25	535	17.10	600	20.0	20.10	30.85	47.7	.0647	30.0	67.8	196.5	.908	1538	1503	1472	2027
P _i = 50 Re = 60	4/4	1740	12 30.00	20.55	50.55	9.70	535	16.30	600	20.0	30.20	30.00	46.2	.0644	27.8	62.8	182.1	.874	34.2	33.6	32.8	46.2
"	4/5	1715	6 29.70	20.60	50.30	9.60	536	16.30	599	20.0	30.20	30.00	46.0	.0652	27.4	61.9	174.1	.876	1511	1485	1450	2062

TABLE II.

CALCULATED OUTPUT CEE SYSTEM.

AIR.	P ₁	P ₂	T _e	W _c	W _t	W _e	V _{net}	lbs. Air/hr	H.P. net
S.L.	50	30	1564	23	0	381	358	502	70.6
S.L.	50	40	1564	23	29	369	375	496	73.0
S.L.	50	50	1538	23	49	359	385	480	72.6
S.L.	50	60	1511	23	64	344	385	461	69.6
15,000	50	30	1564	51	55	381	385	502	76.1
15,000	50	40	1564	51	80	369	398	496	78.1
15,000	50	50	1538	51	97	359	405	480	77.0
15,000	50	60	1511	51	109	344	402	461	73.4
30,000	50	30	1564	75	108	381	414	502	81.6
30,000	50	40	1564	75	129	369	423	496	82.5
30,000	50	50	1538	75	143	359	427	480	80.5
30,000	50	60	1511	75	152	344	421	461	76.1
S.L.	40	30	1529	11	0	374	363	392	55.8
S.L.	40	40	1511	11	28	364	381	374	56.0
S.L.	40	50	1476	11	48	354	391	360	55.3
S.L.	40	60	1459	11	62	335	386	348	52.8
15,000	40	30	1529	38	54	374	390	392	60.3
15,000	40	40	1511	38	78	364	404	374	59.6
15,000	40	50	1476	38	94	354	410	360	59.8
15,000	40	60	1459	38	106	335	403	348	55.4
30,000	40	30	1529	63	106	374	417	392	64.2
30,000	40	40	1511	63	126	364	427	374	62.8
30,000	40	50	1476	63	138	354	429	360	60.8
30,000	40	60	1459	63	148	335	420	348	57.4

T_e = exhaust temperature, °F, used in calculation of turbine work.

W_t = turbine work, BTU/lb. air.

W_c = compressor work, BTU/lb. air.

W_e = engine brake work, BTU/lb. air.

25

[illegible]

MASS RATE OF AIRFLOW VS. EXHAUST PRESSURE

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

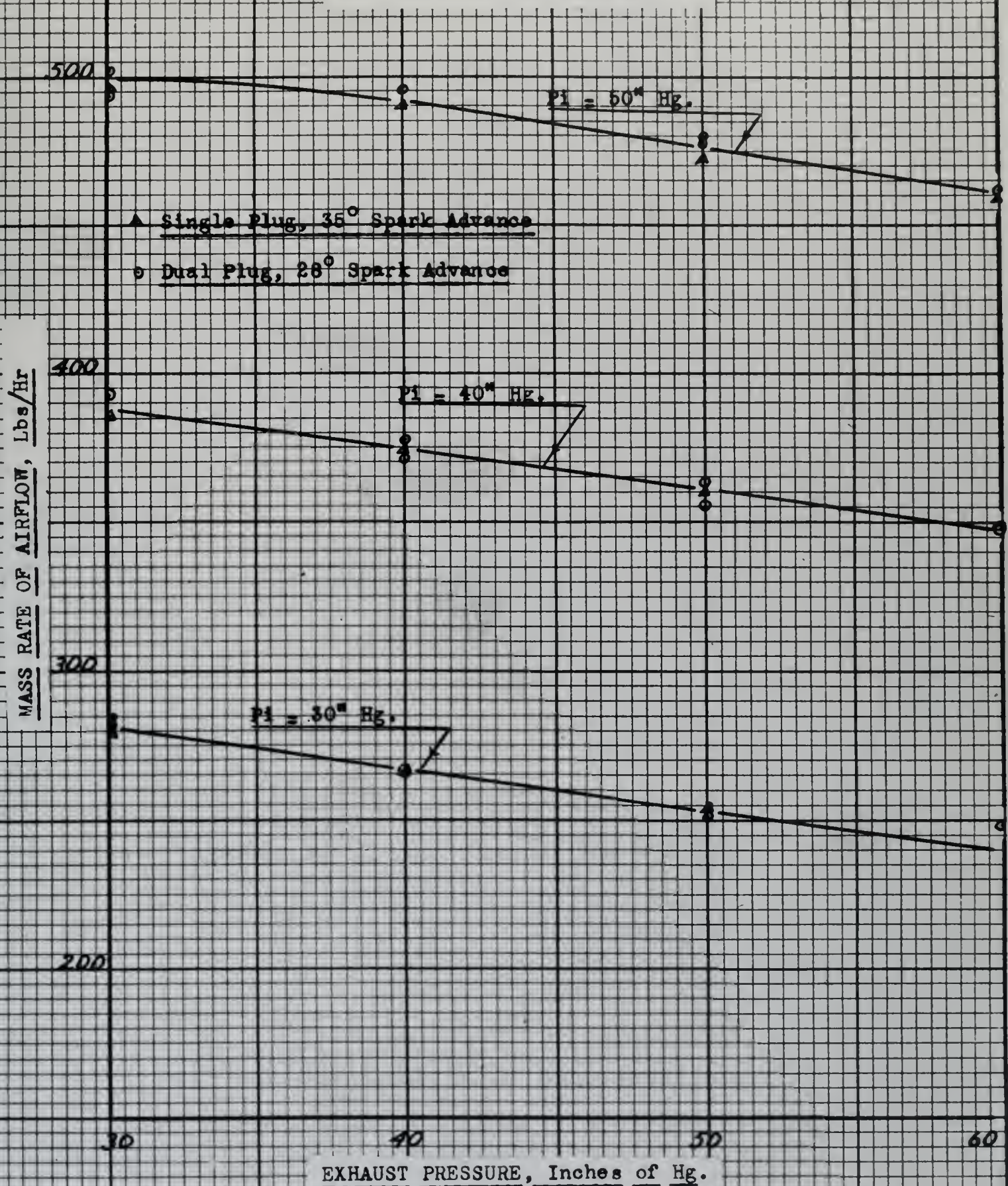
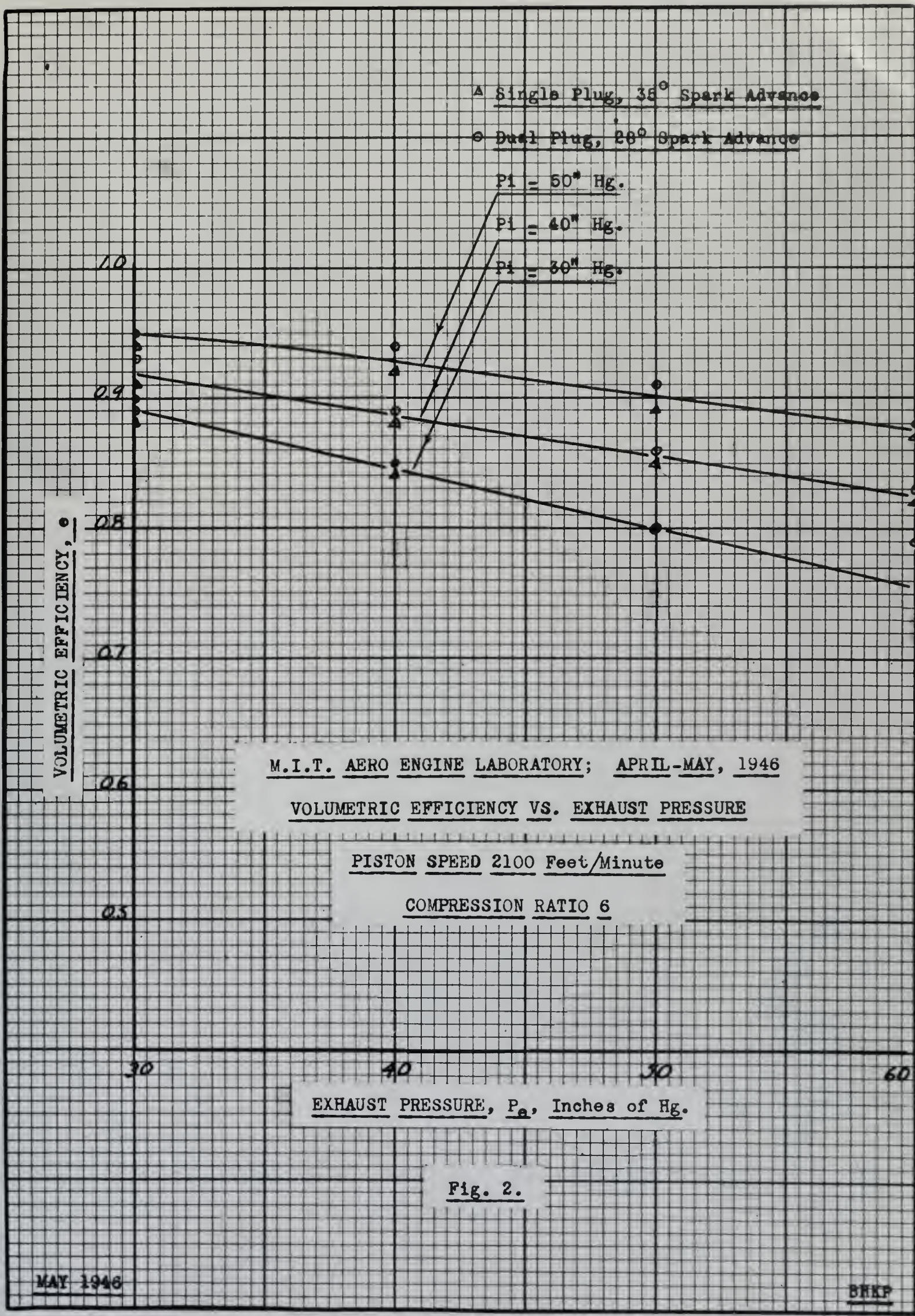


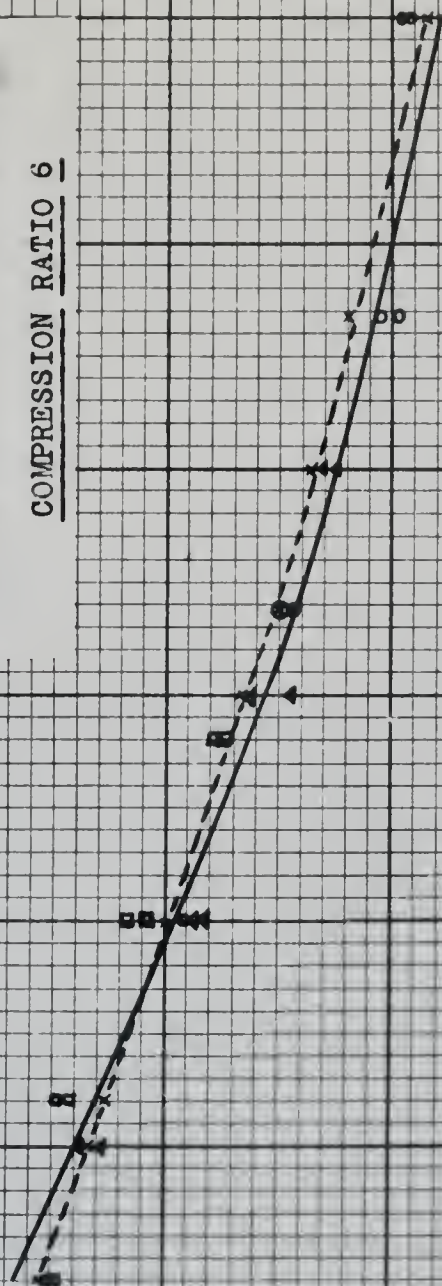
Fig. 1.



VOLUMETRIC EFFICIENCY VS. PRESSURE RATIO

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



THEORETICAL

ACTUAL

x = THEORETICAL
o = ACTUAL, $P_1 = 30''$ Hg.
Δ = ACTUAL, $P_1 = 40''$ Hg.
□ = ACTUAL, $P_1 = 50''$ Hg.

PRESSURE RATIO, P_0/P_1

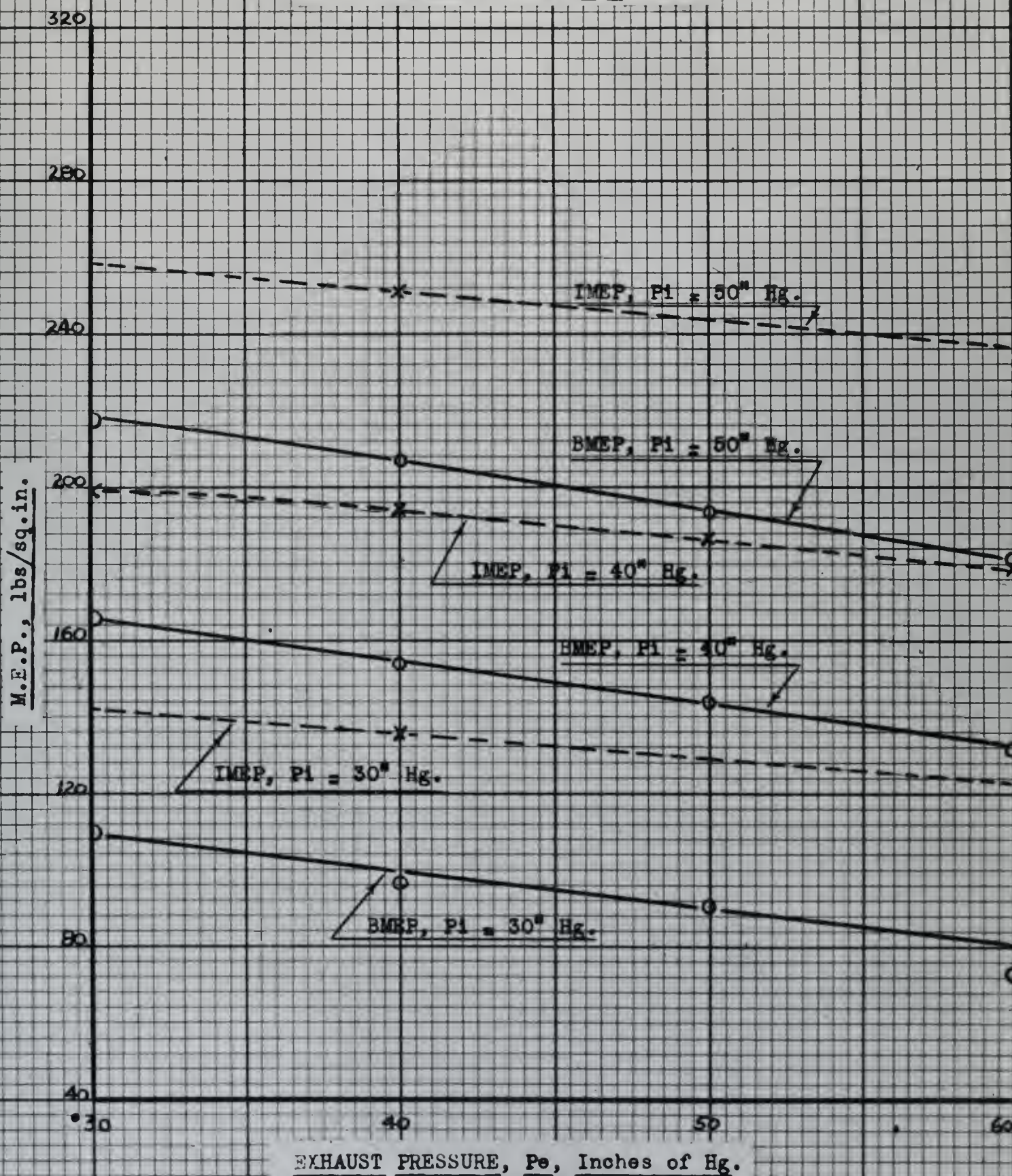
Fig. 3.

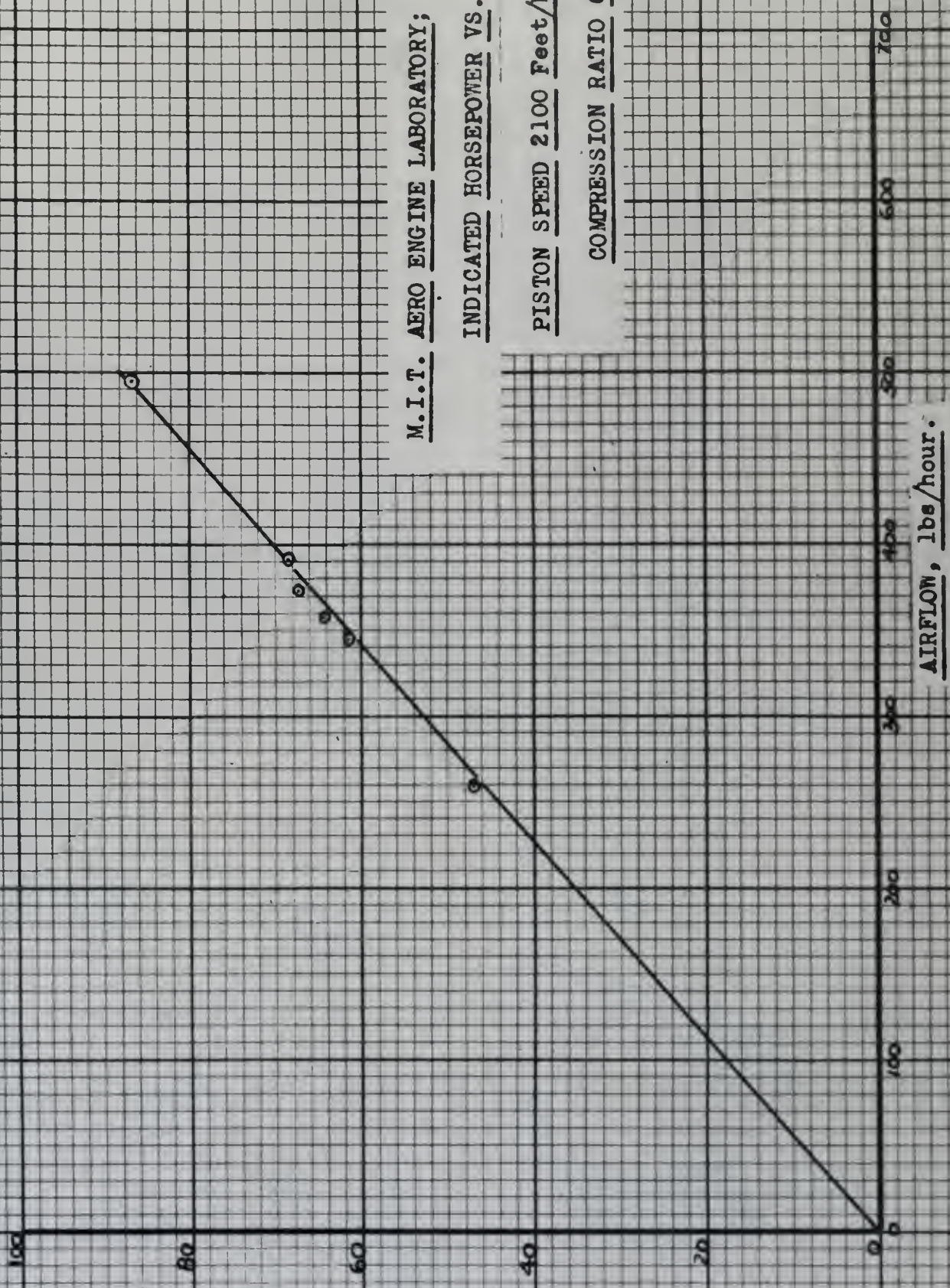
M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

M.E.P. VS. EXHAUST PRESSURE

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6





I.H.P.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

INDICATED HORSEPOWER VS. AIRFLOW

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

AIRFLOW, lbs/hour.

Fig. 5.

MAY 1946

BRAP

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

M.E.P. VS. EXHAUST PRESSURE WITH INLET
PRESSURE CONSTANT AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

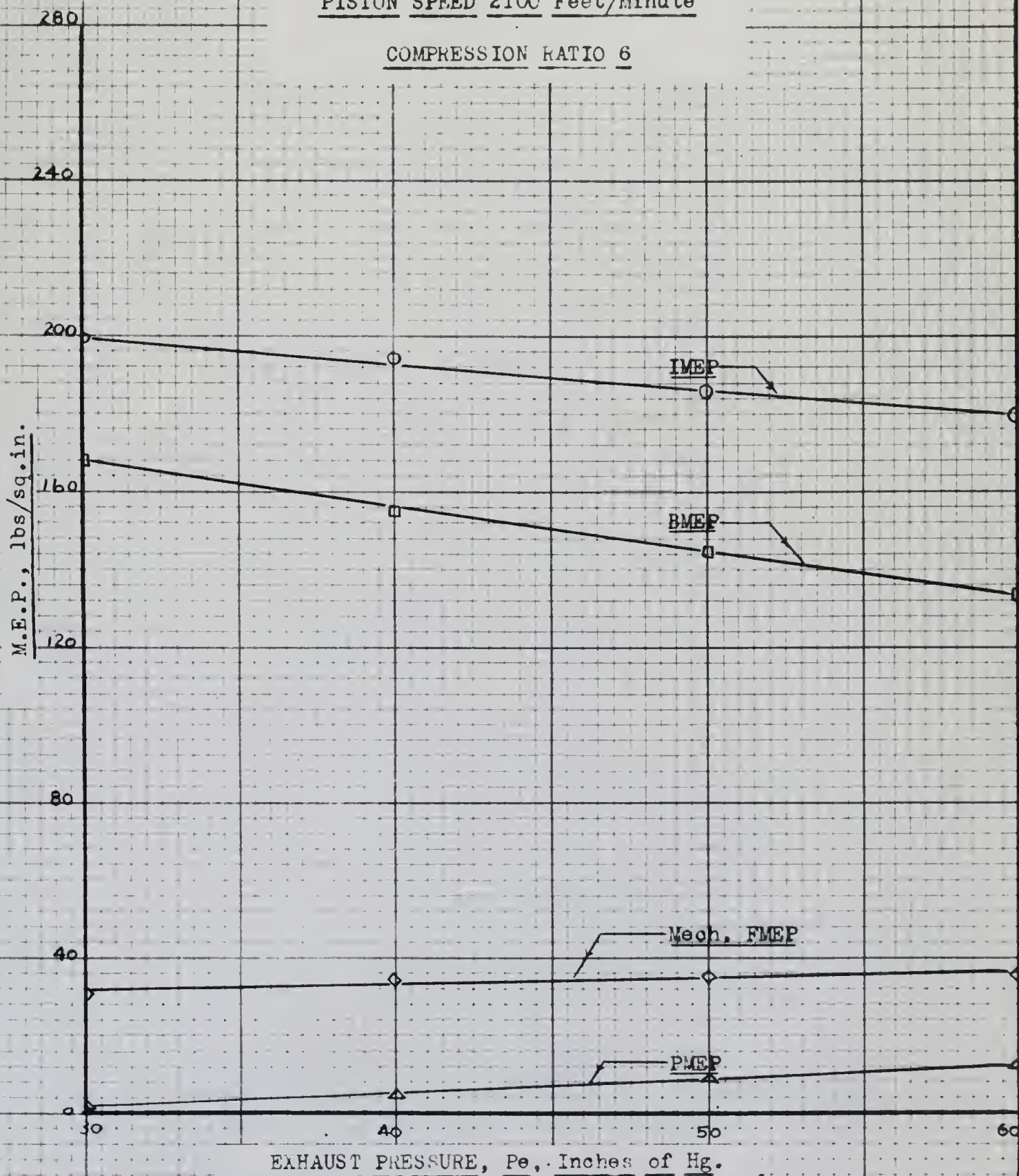


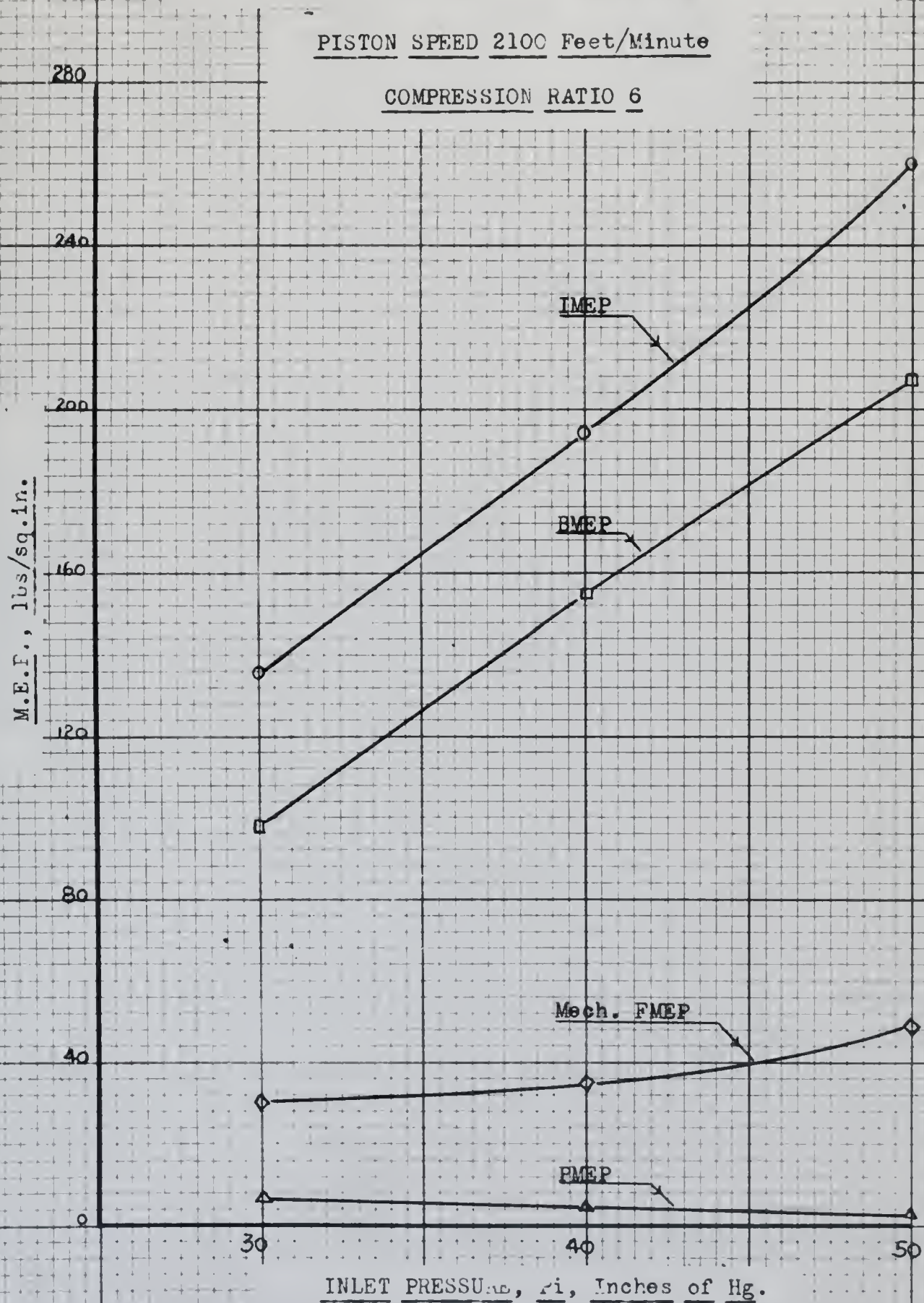
Fig. 6.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

M.E.P. VS. INLET PRESSURE WITH EXHAUST
PRESSURE CONSTANT AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



MAY 1946

Fig. 7.

BHCP

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

BRAKE HORSEPOWER VS. EXHAUST PRESSURE

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

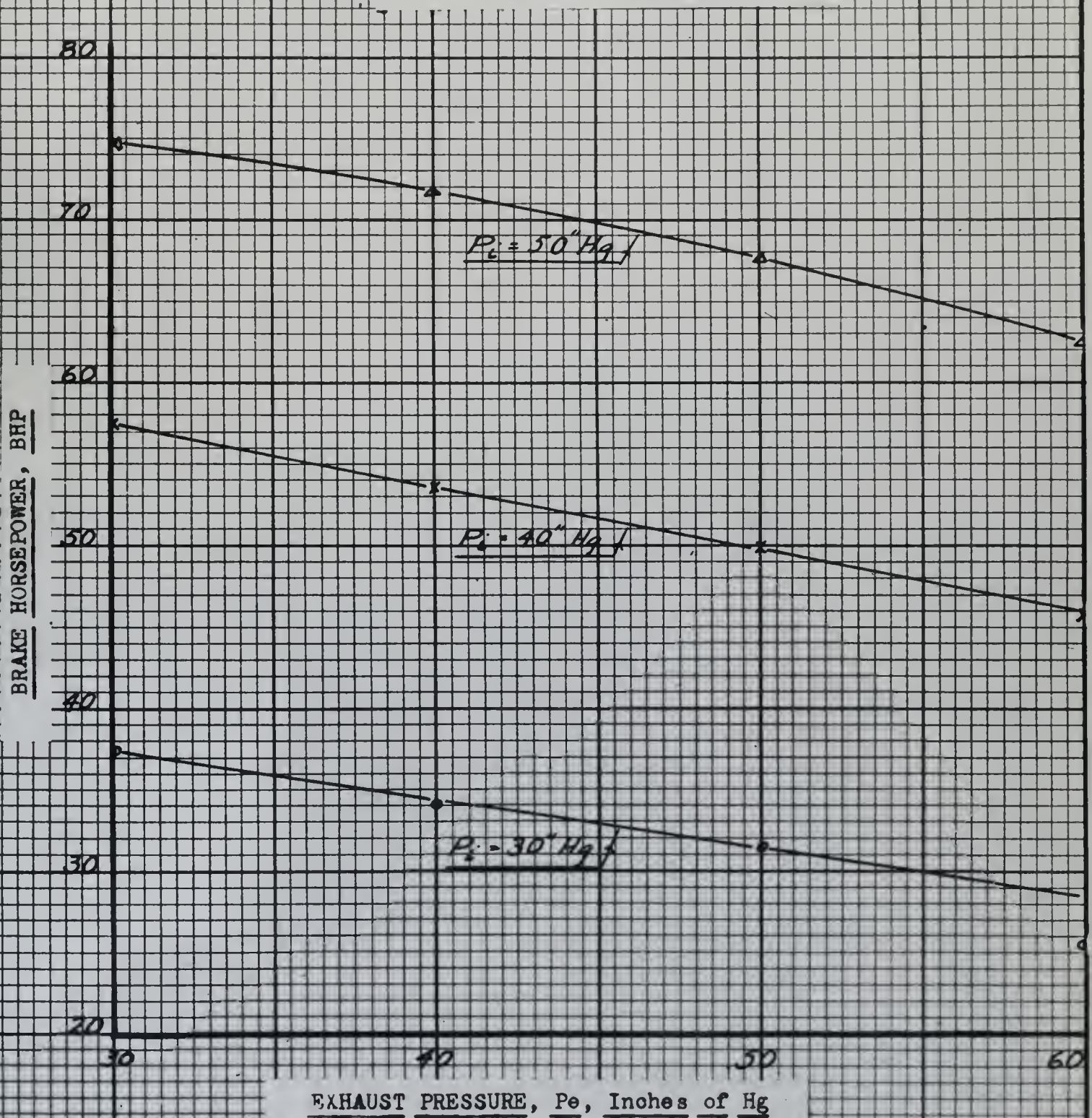


Fig. 8.

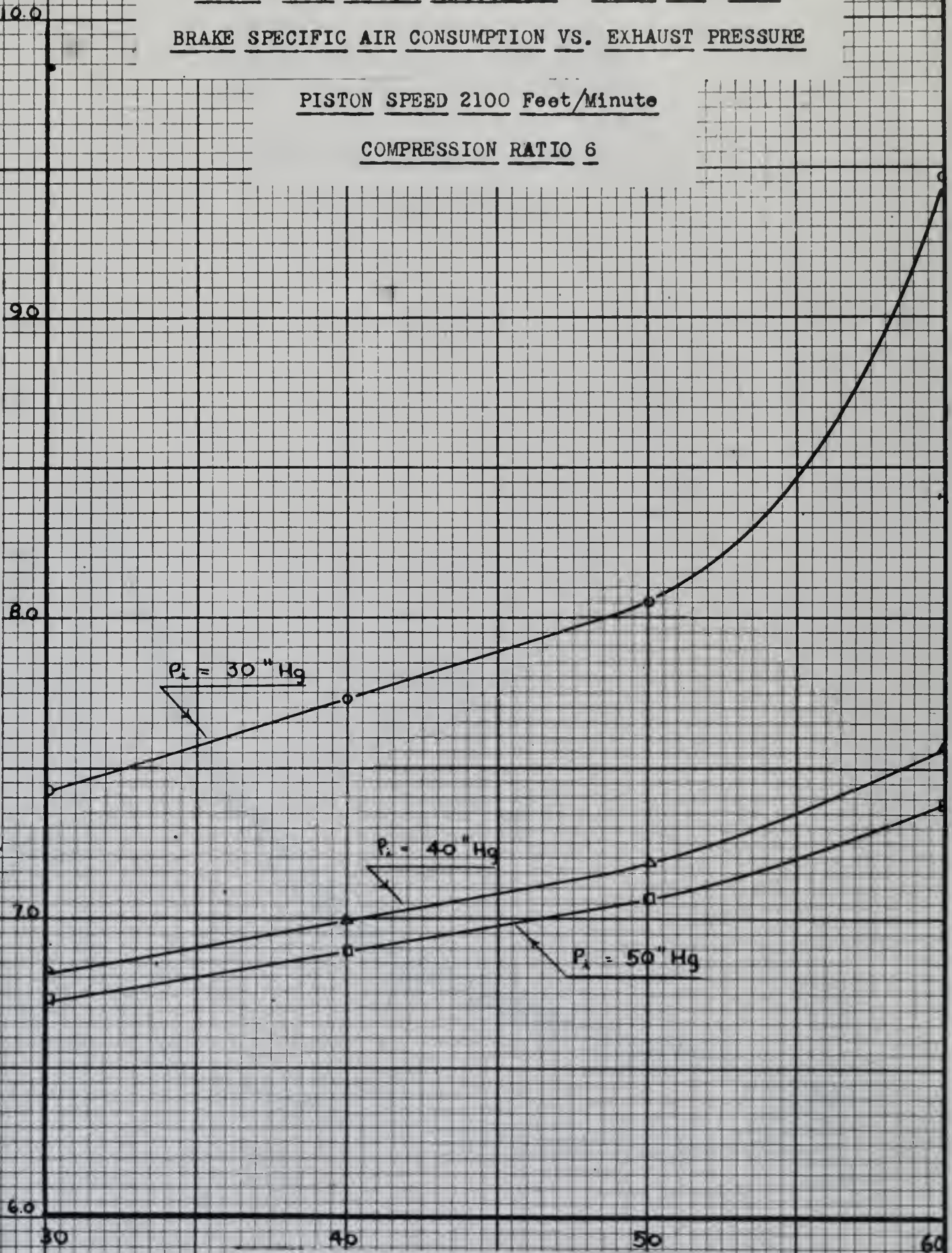
M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

BRAKE SPECIFIC AIR CONSUMPTION VS. EXHAUST PRESSURE

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

BRAKE SPECIFIC AIR CONSUMPTION, Lbs/BHP-hr



EXHAUST PRESSURE, P_e , Inches of Hg.

MAY 1946

Fig. 9.

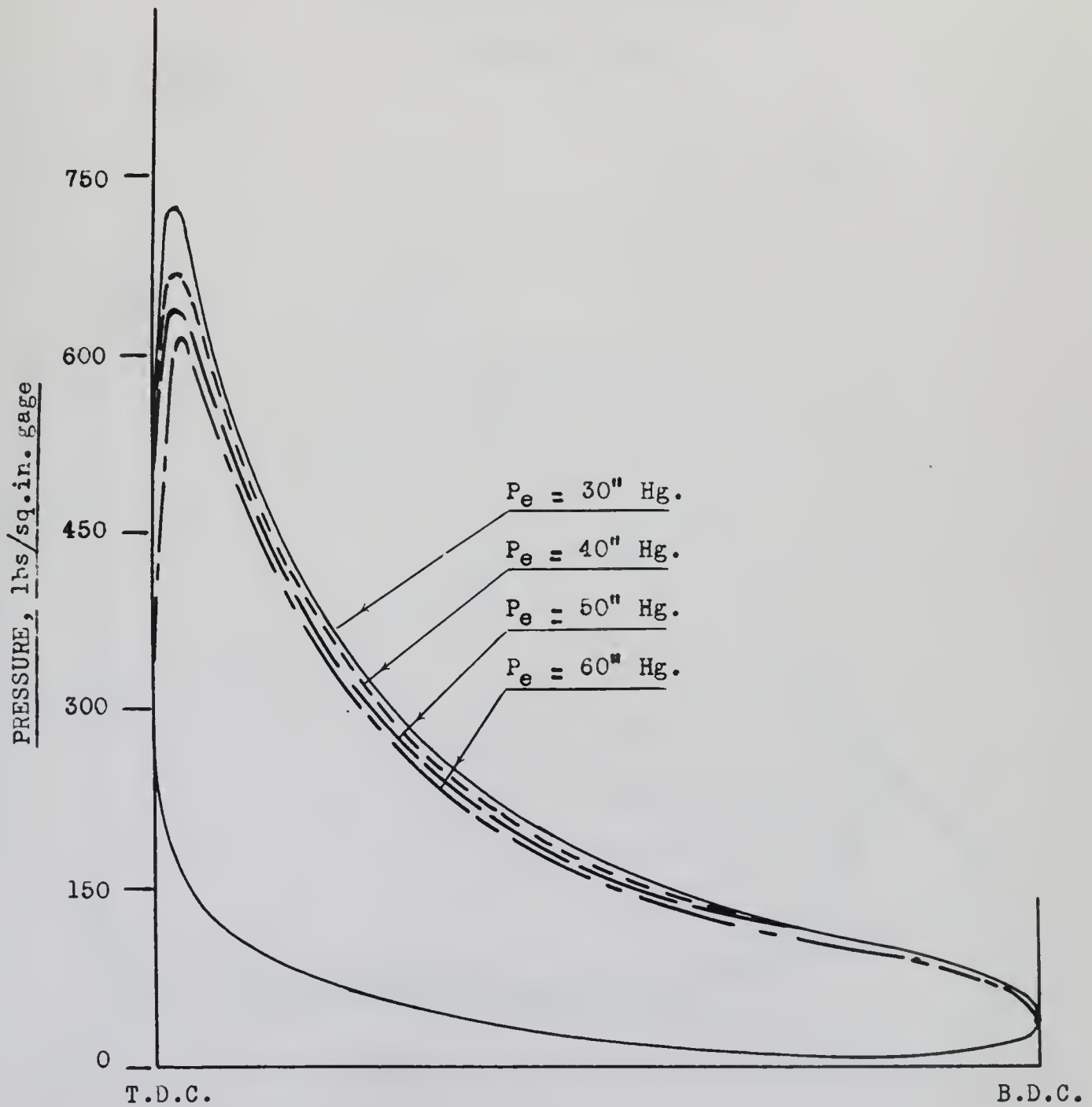
BHKB

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EFFECT OF EXHAUST PRESSURE ON
THE INDICATOR CARD WITHOUT PUMPING
LOOP WITH INLET PRESSURE CONSTANT
AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



MAY 1946

Fig. 10.

BHKP

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EFFECT OF EXHAUST PRESSURE ON
THE PUMPING DIAGRAM WITH INLET
PRESSURE CONSTANT AT 40" Hg.

BORE STROKE ENGINE

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

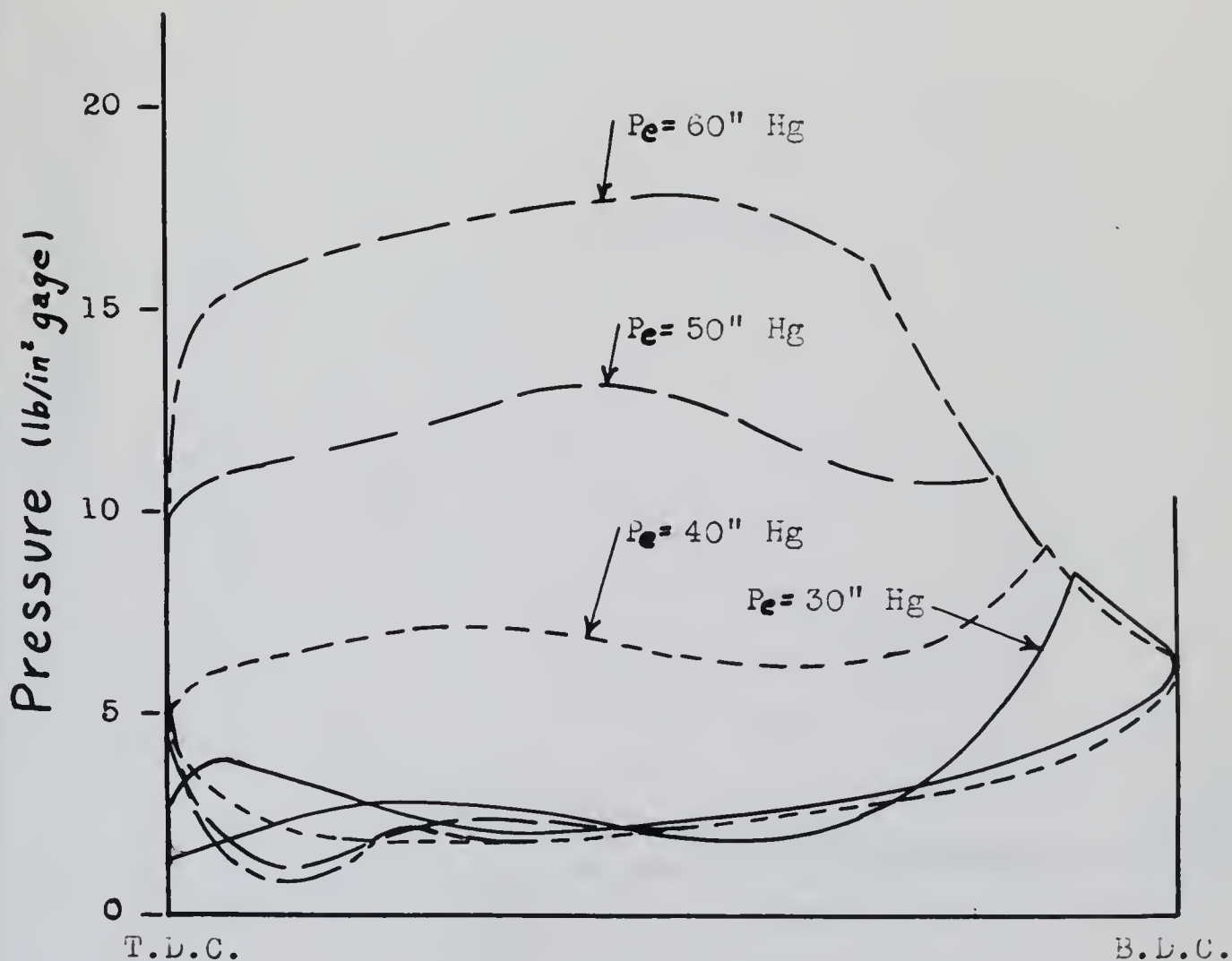


Fig. 11.

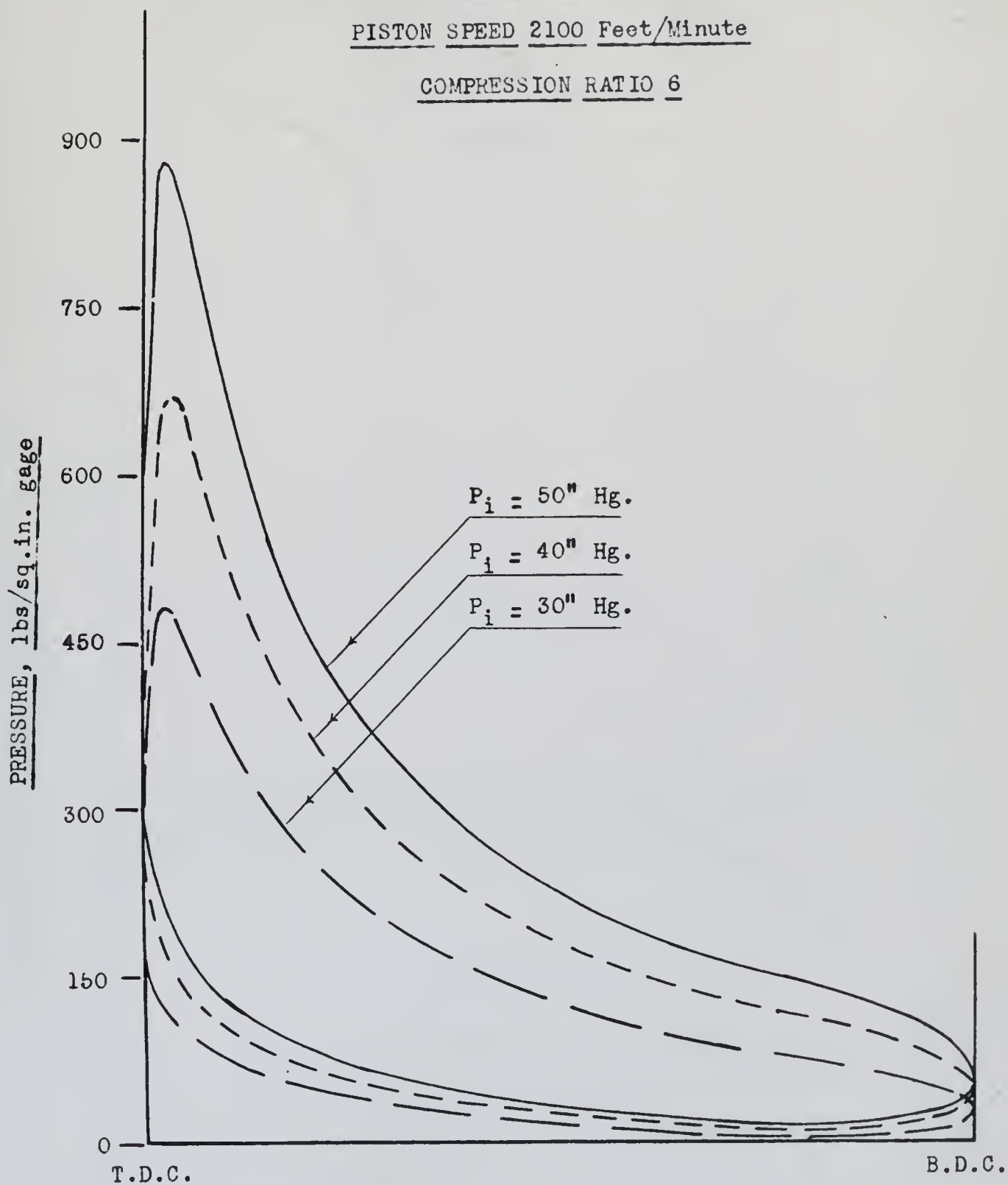
BHKP

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EFFECT OF INLET PRESSURE ON THE
INDICATOR CARD WITHOUT PUMPING
LOOP WITH EXHAUST PRESSURE CONSTANT
AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



MAY 1946

BHKP

Fig. 12.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EFFECT OF INLET PRESSURE ON THE
PUMPING DIAGRAM WITH EXHAUST
PRESSURE CONSTANT AT 40" Hg.

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

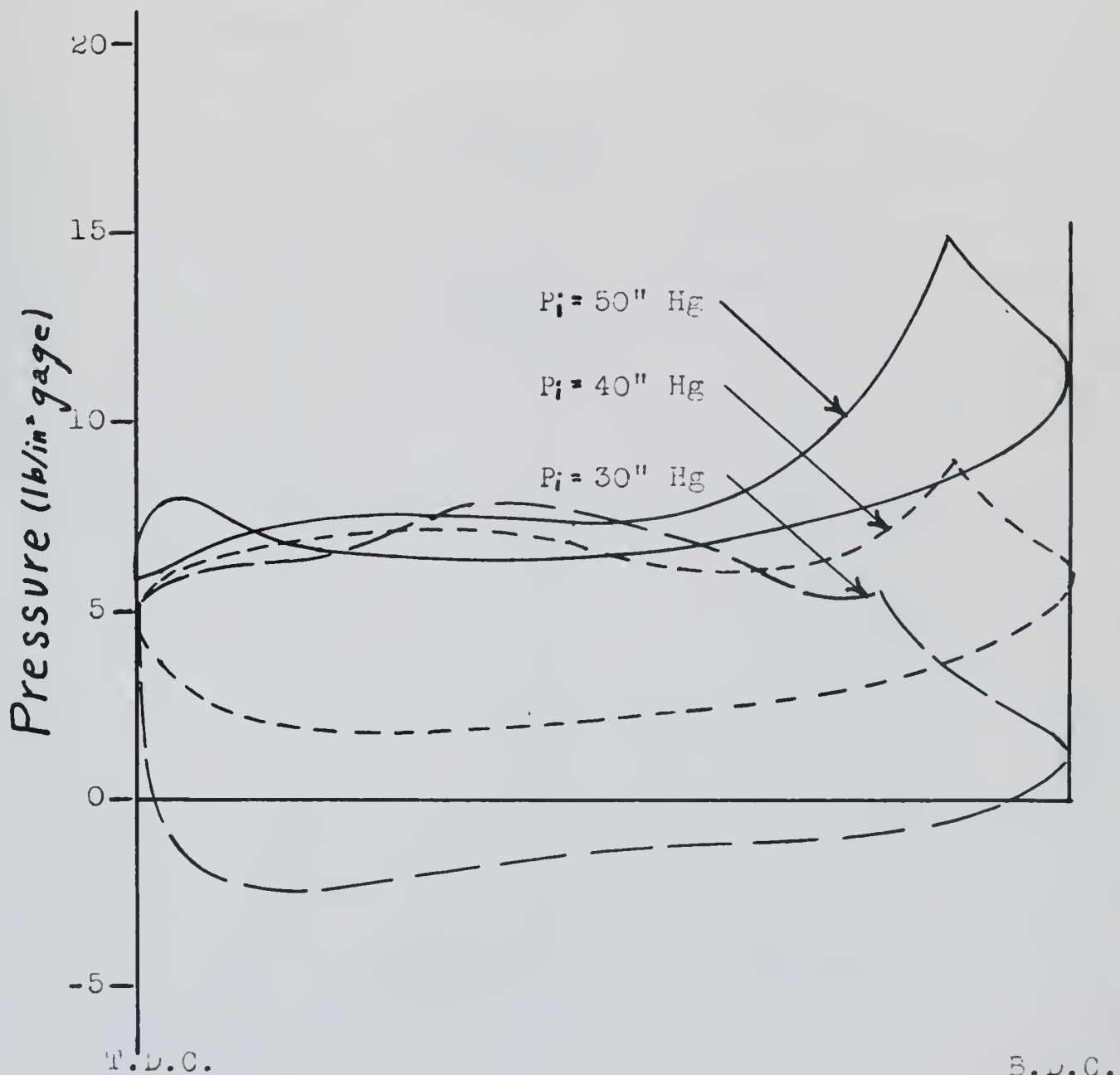


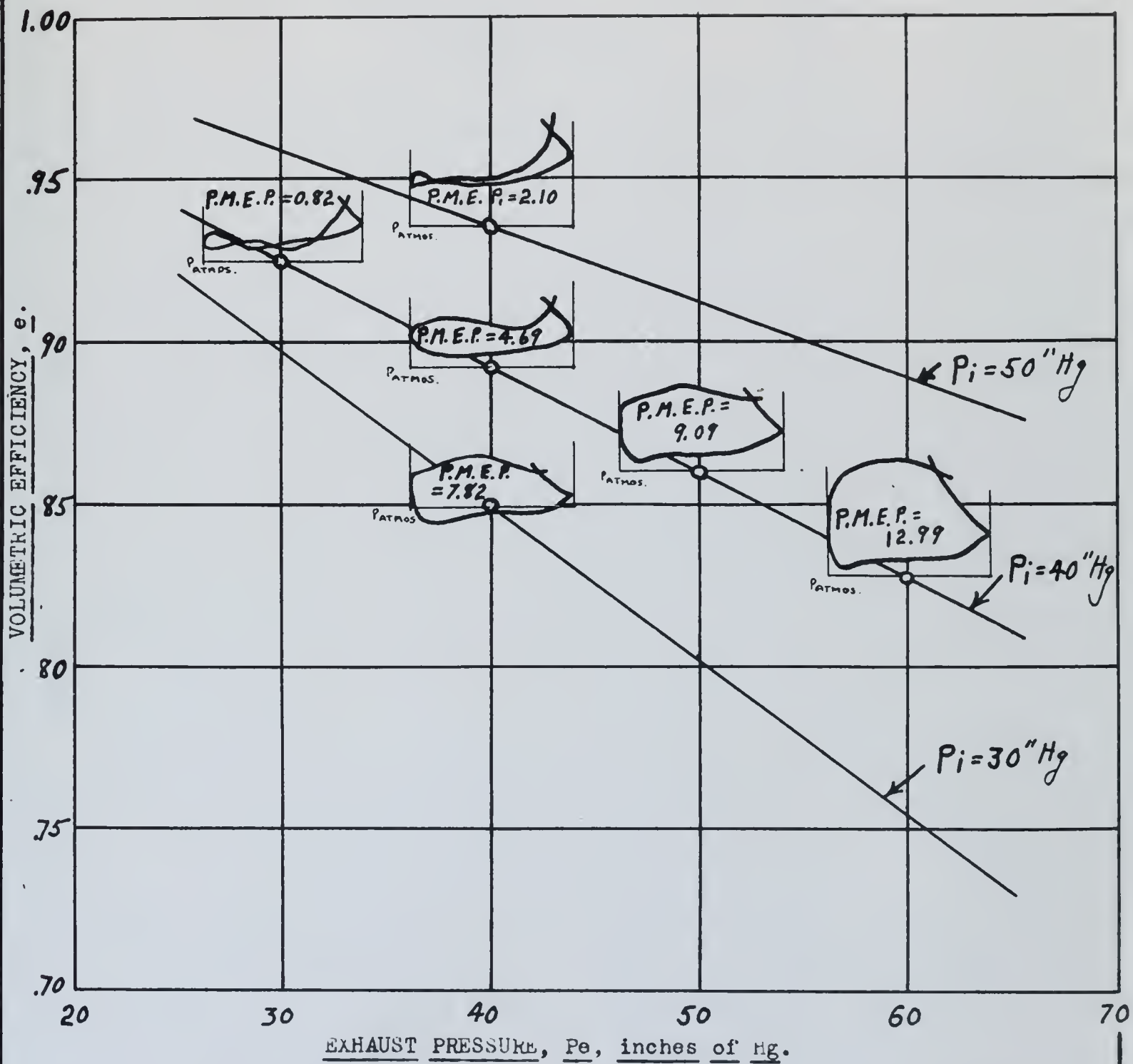
Fig. 13.

BHKP

RELATION OF PUMPING CYCLE TO
VOLUMETRIC EFFICIENCY

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6



BHCP

Fig. 14.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EXHAUST GAS TEMPERATURES VS. AMOUNT
OF OPENING OF BLEED VALVE

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

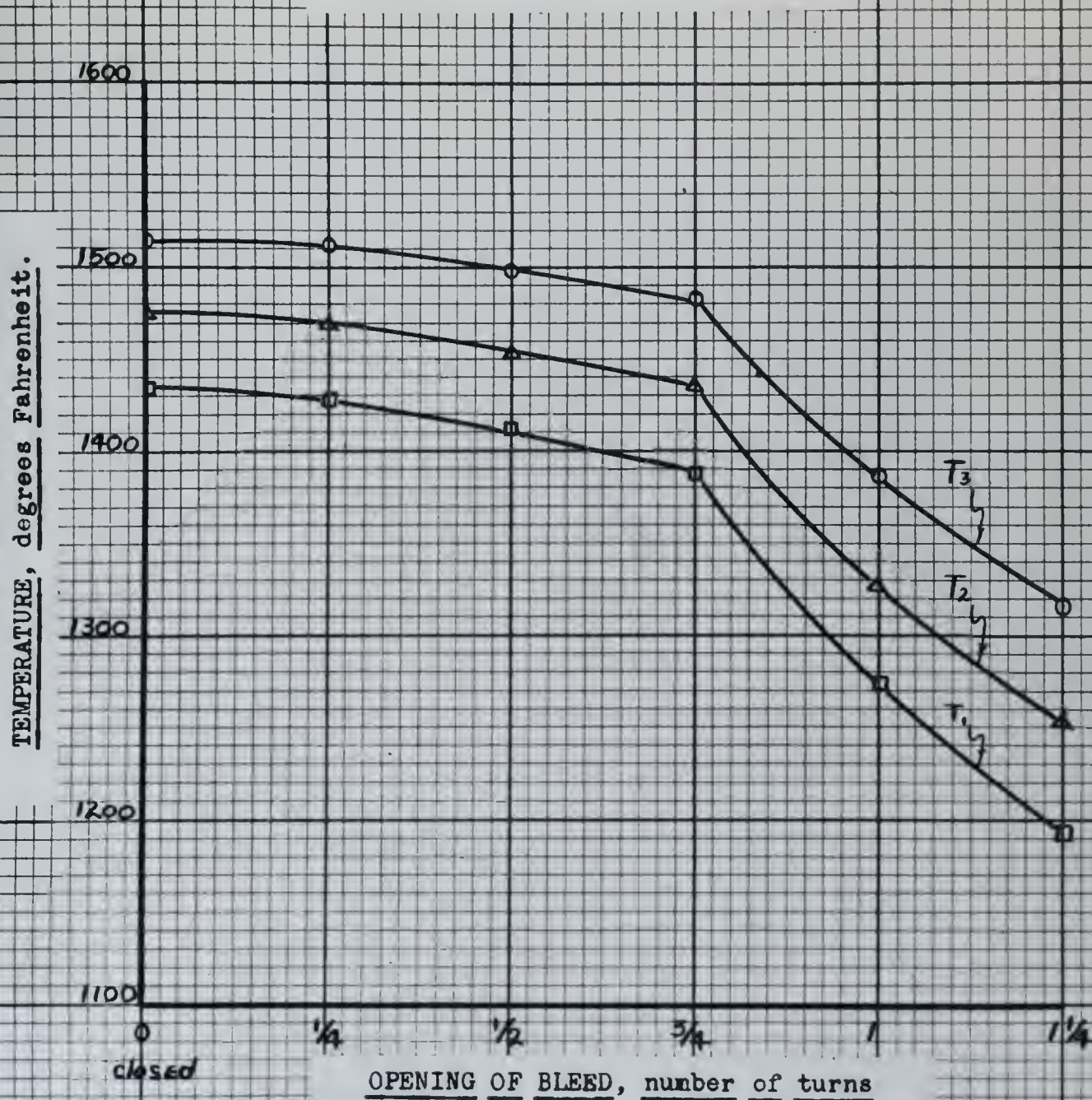


Fig. 15.

20000

1900

T_4 , Temperatures in Exhaust Stack
(For all values of inlet pressure)

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

EXHAUST TEMPERATURE VS. EXHAUST PRESSURE

1800

PISTON SPEED 2100 Feet/Minute

COMPRESSION RATIO 6

1700

1600

T_3 , Temperatures in Calorimeter

$P_1 = 50''$ Hg.

$P_1 = 40''$ Hg.

$P_1 = 30''$ Hg.

1500

1400

1300

1200

30

40

50

60

EXHAUST PRESSURE, P_e , Inches of Hg.

TEMPERATURE, degrees Fahrenheit

MAY 1946

Fig. 16.

BHCP

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

NET HORSEPOWER FOR
COMPRESSOR-ENGINE-TURBINE SYSTEM
VS. EXHAUST PRESSURE

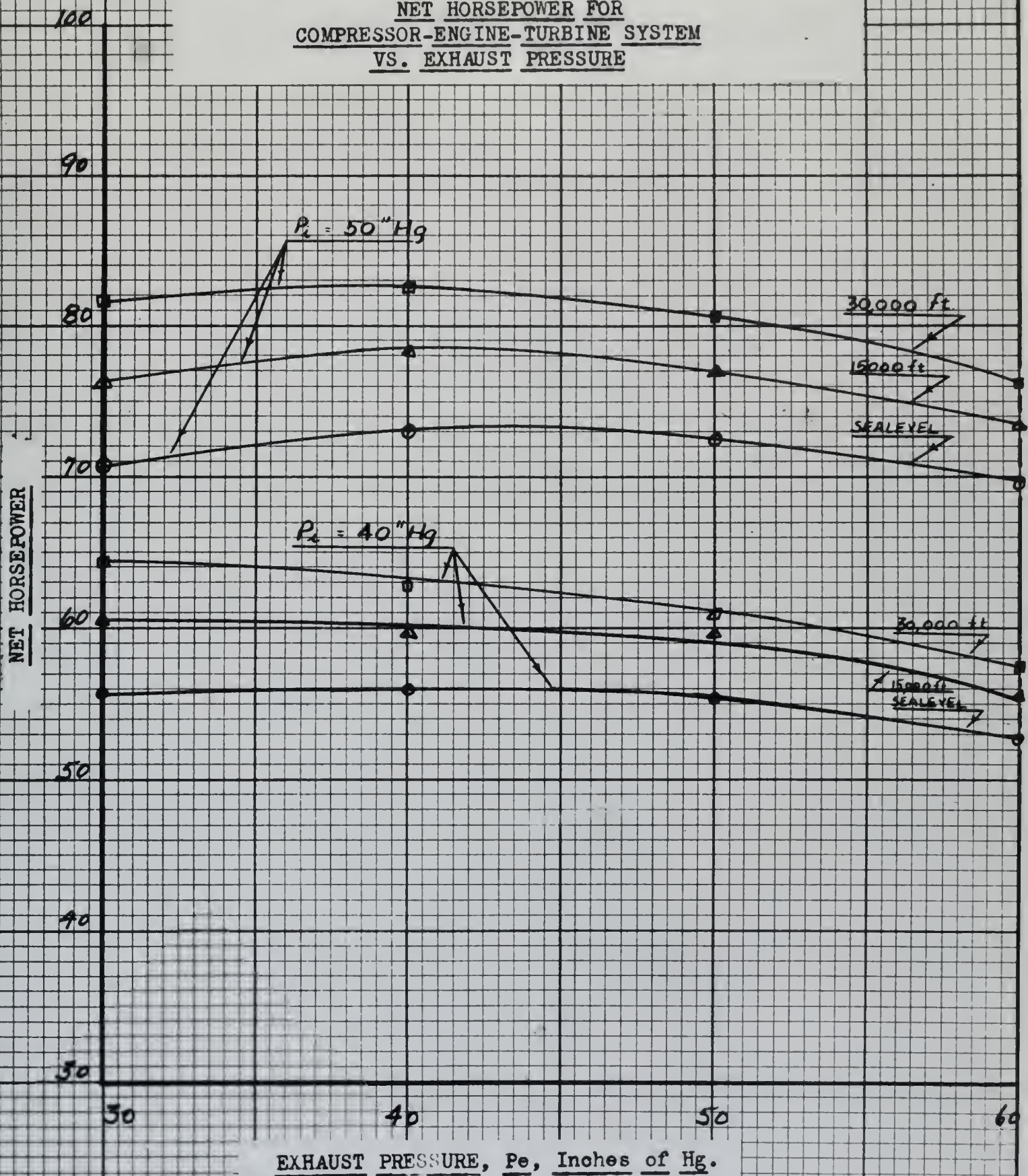


Fig. 17.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

NET HORSEPOWER VS. RATIO OF
EXHAUST PRESSURE/INLET PRESSURE FOR CET SYSTEM

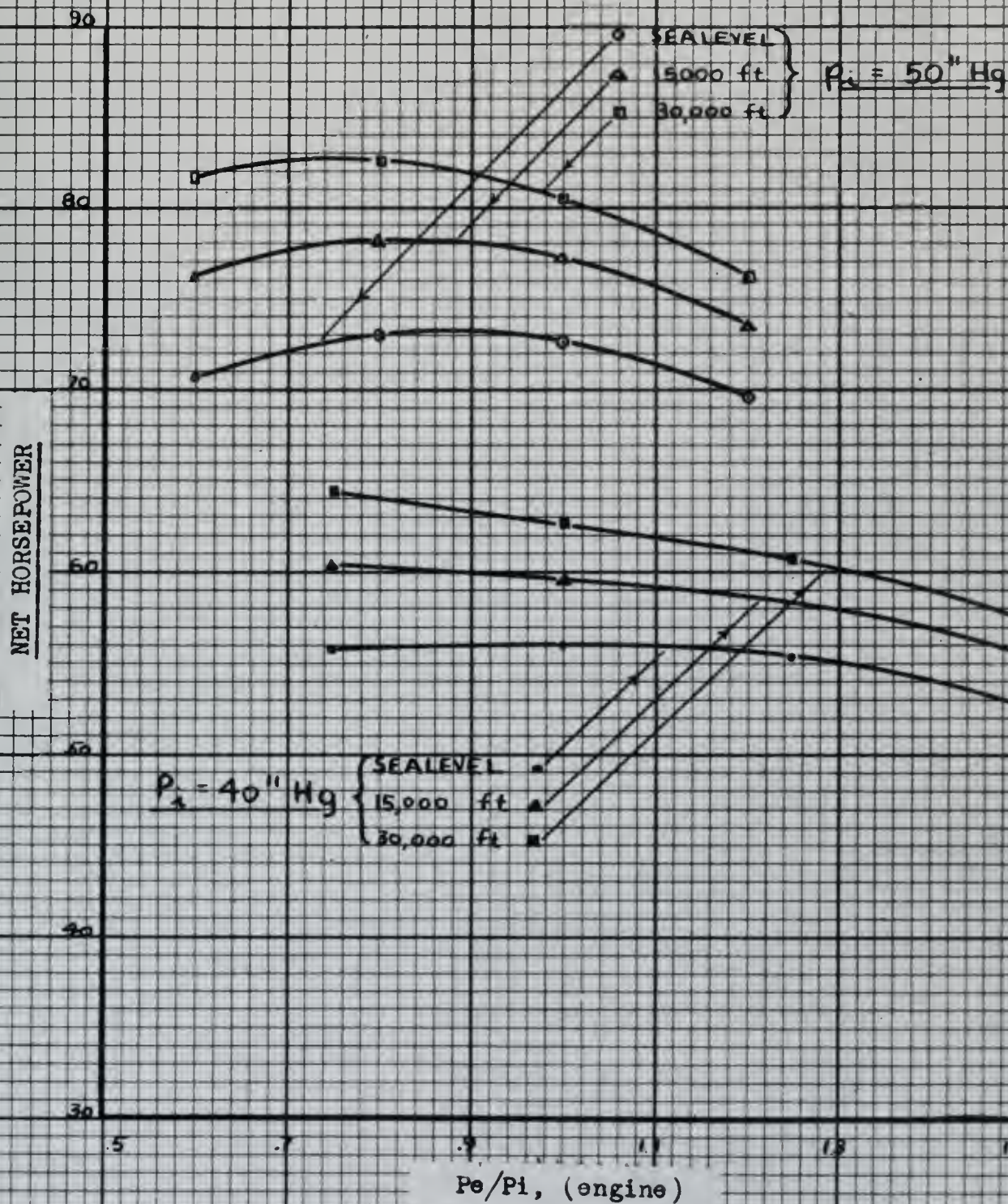


Fig. 18.

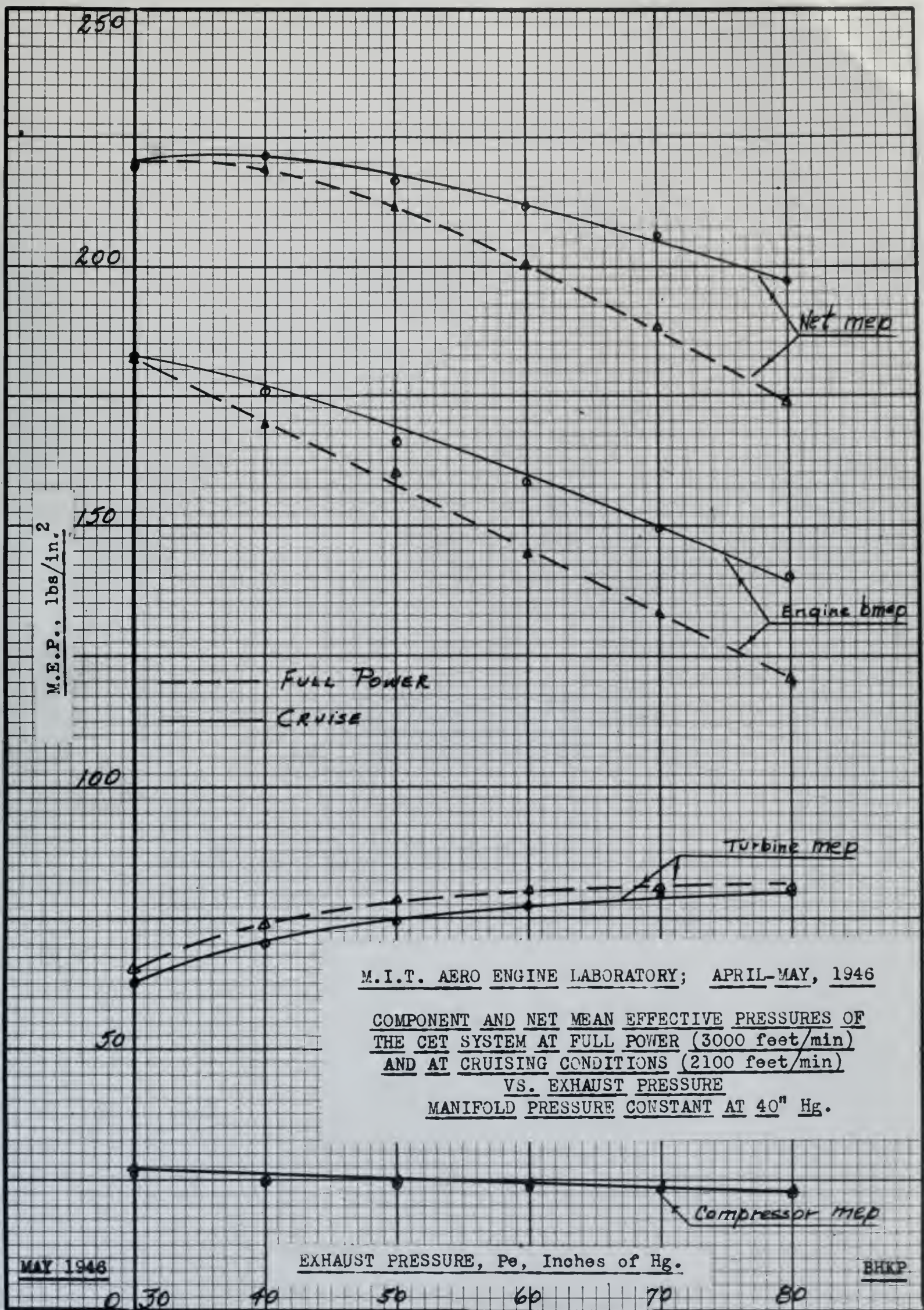


Fig. 19.

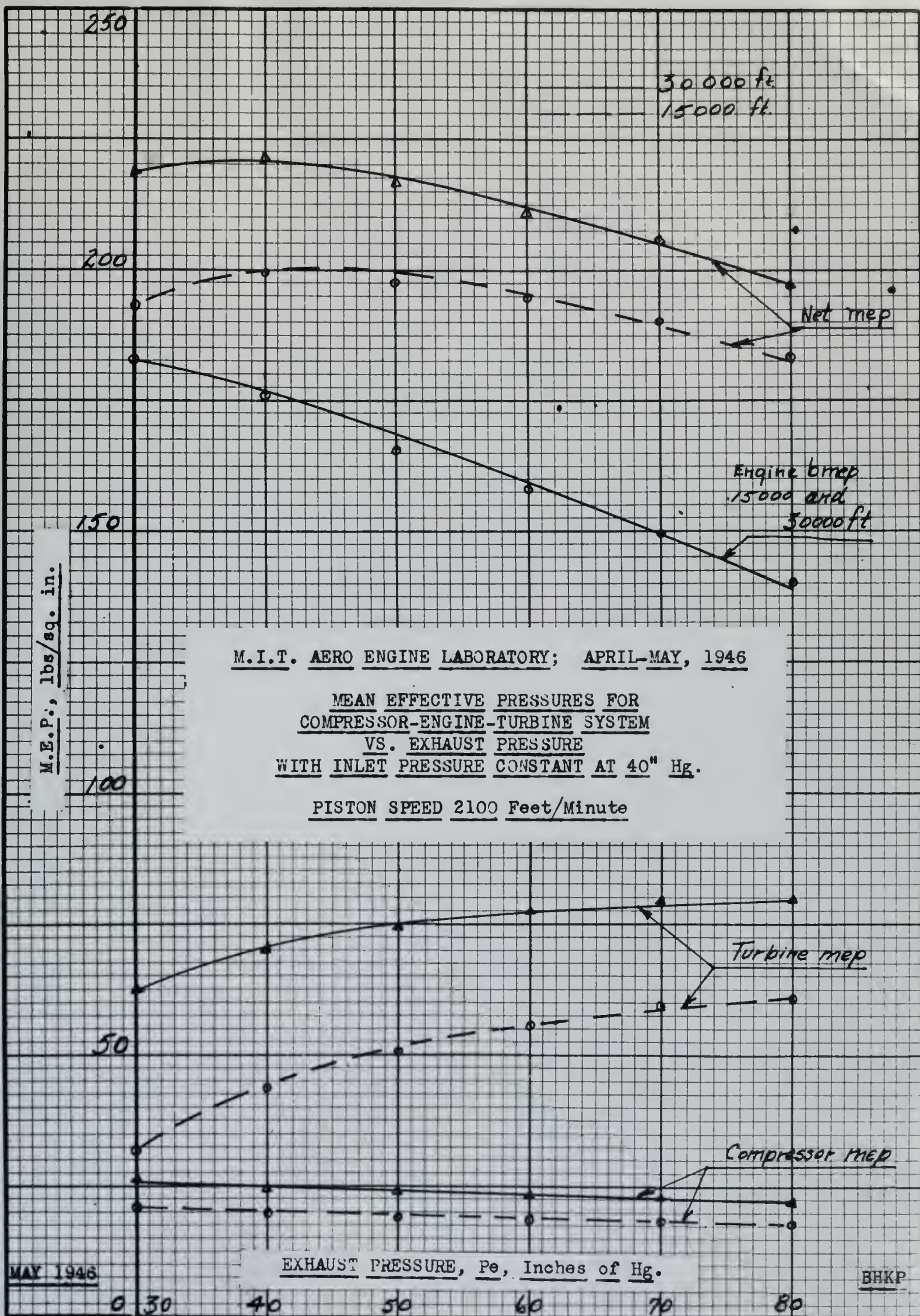
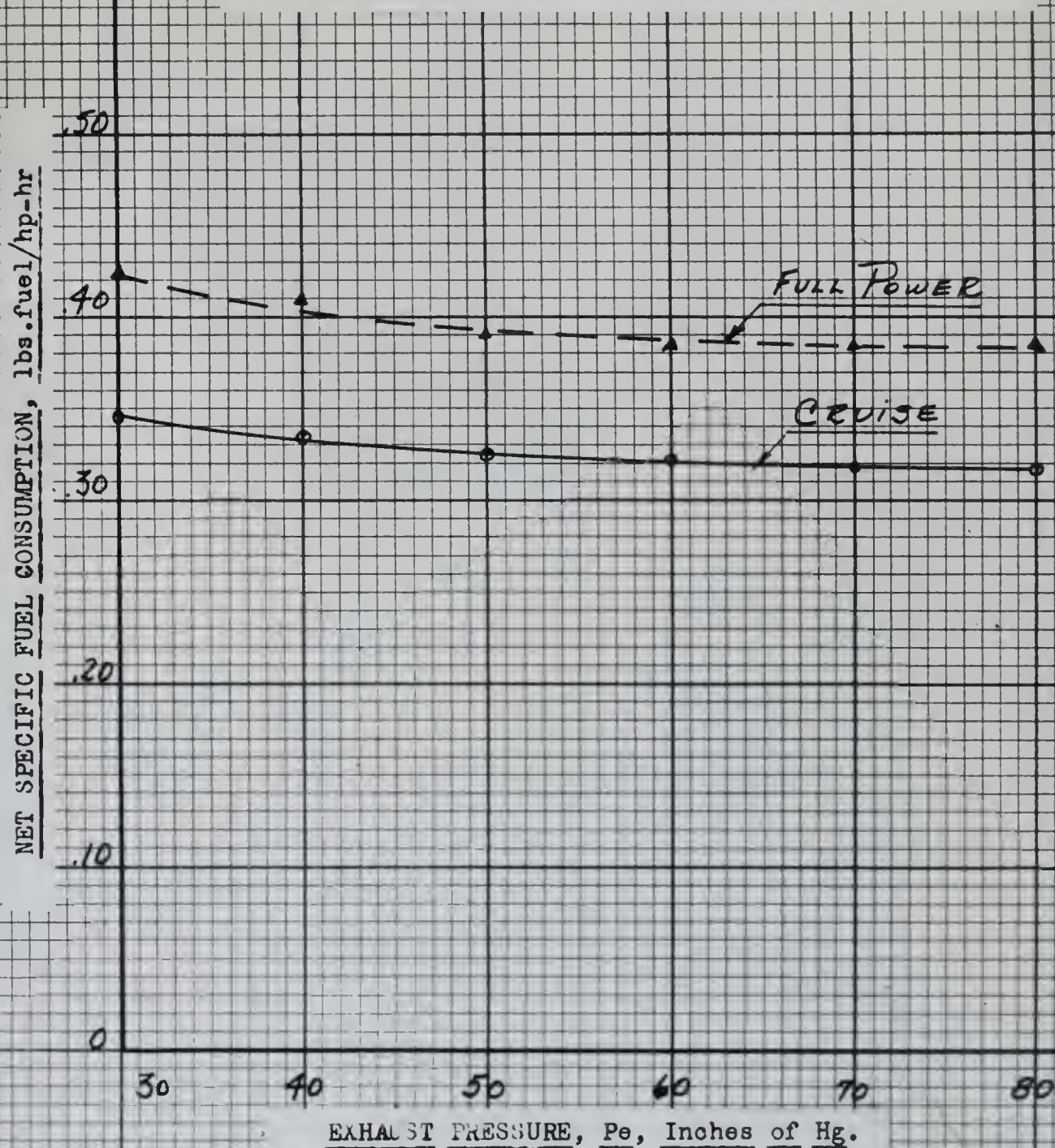


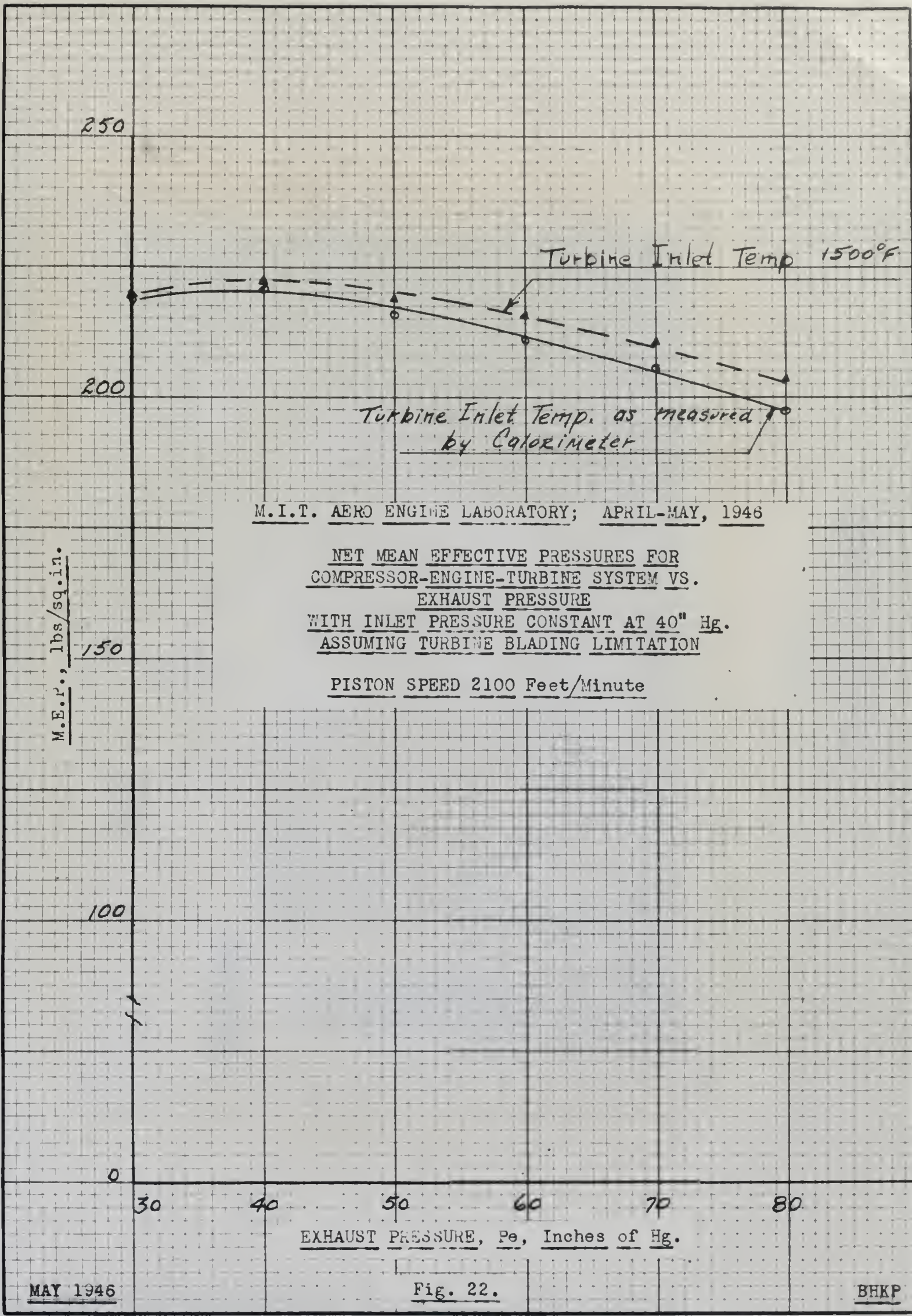
Fig. 20.

M.I.T. AERO ENGINE LABORATORY; APRIL-MAY, 1946

NET SPECIFIC FUEL CONSUMPTION
OF THE CET SYSTEM AT FULL POWER
(3000 feet/min) AND CRUISING
CONDITIONS (2100 feet/min)

VS. EXHAUST PRESSURE
WITH INLET PRESSURE CONSTANT AT 40" Hg.





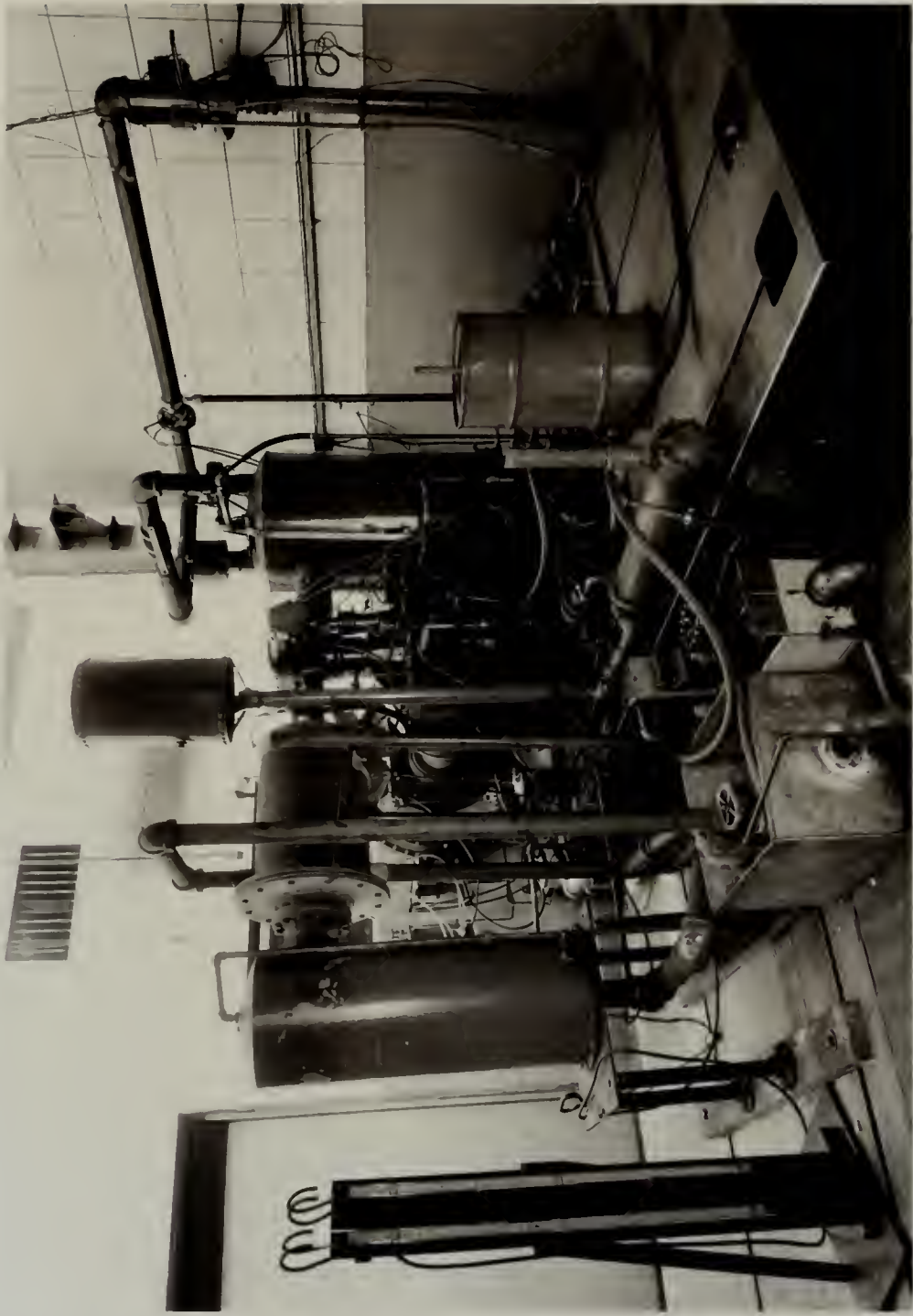


Fig. A.
ARRANGEMENT OF APPARATUS

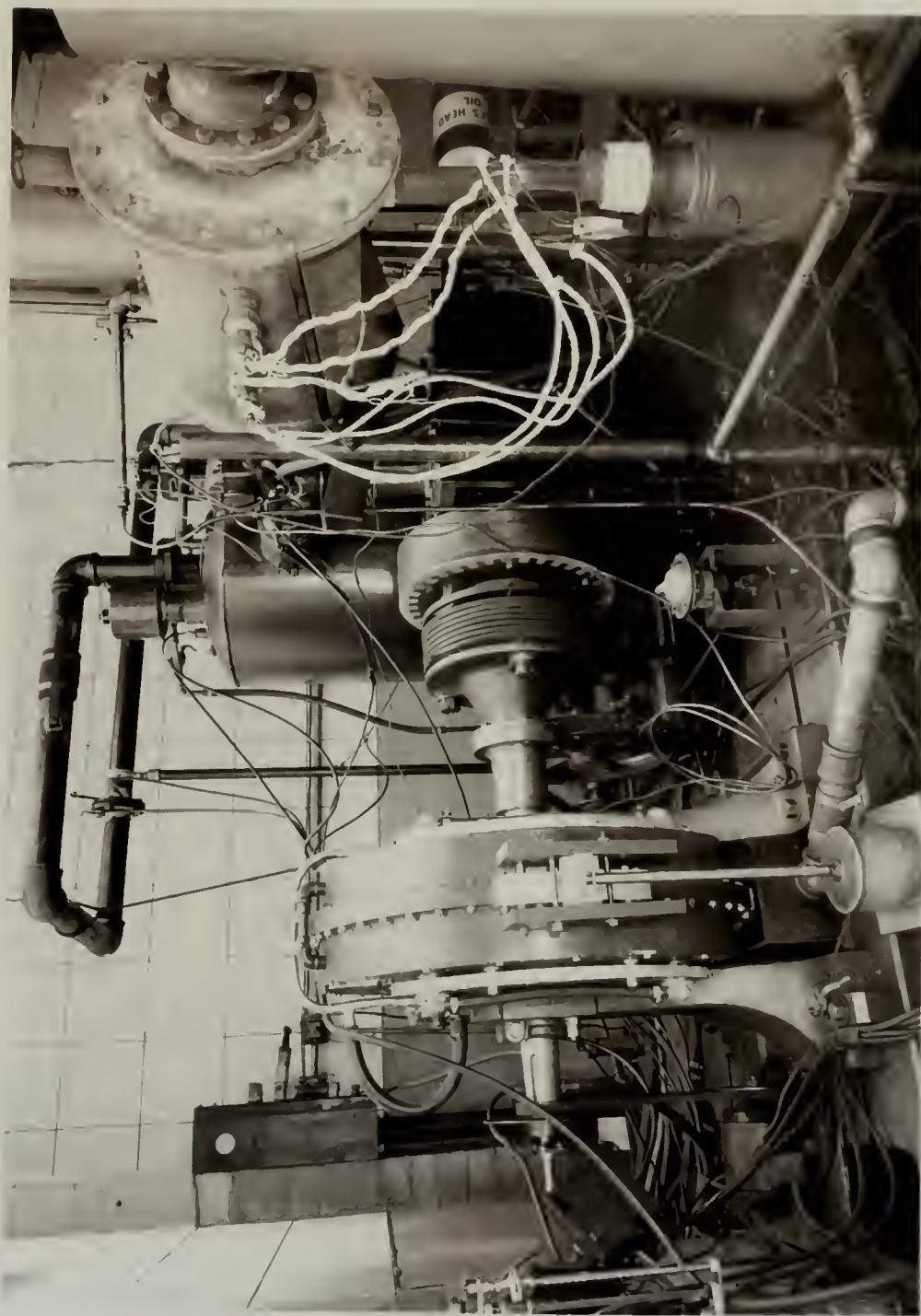


Fig. B
ARRANGEMENT OF APPARATUS



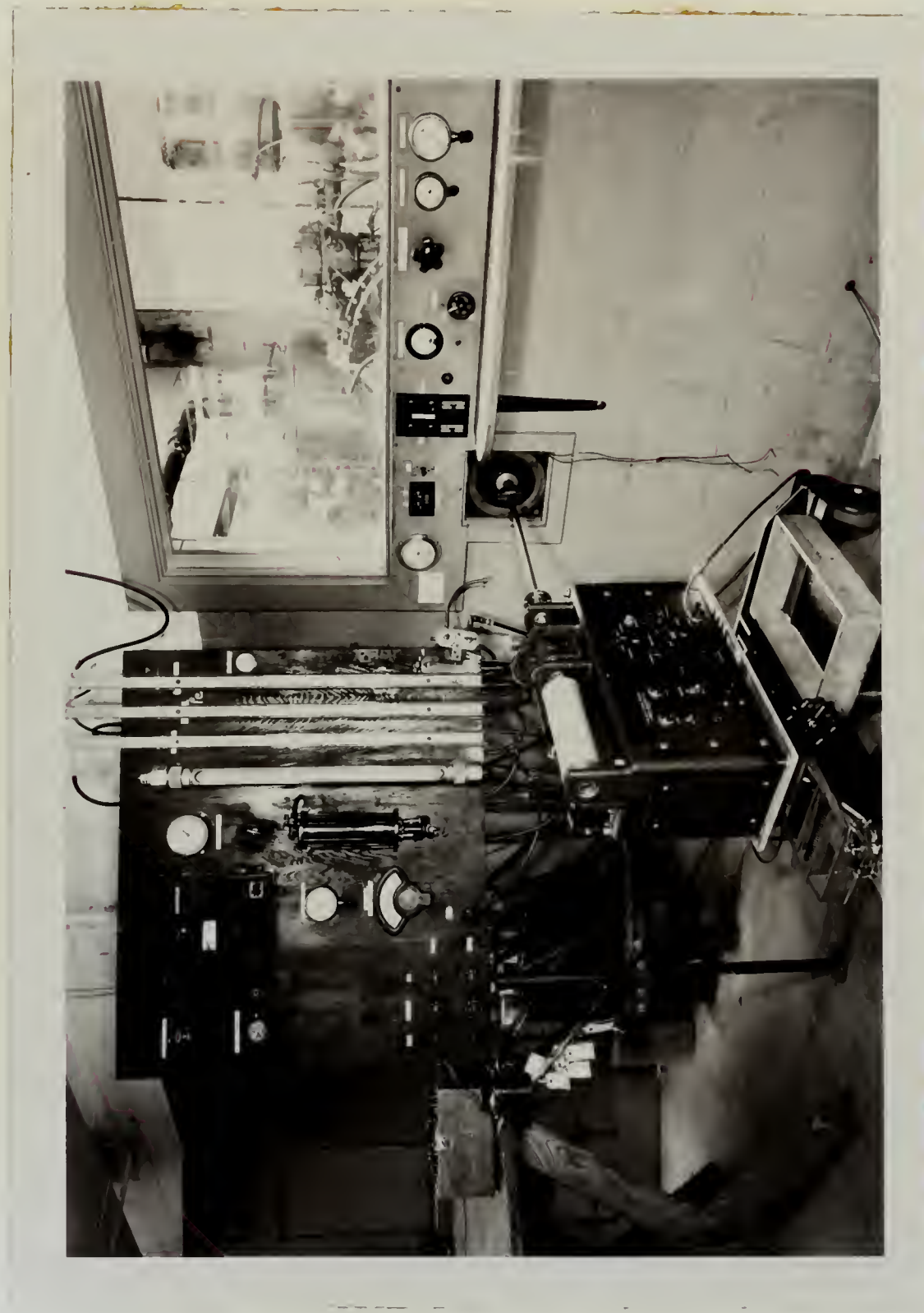


Fig. C
ARRANGEMENT OF APPARATUS



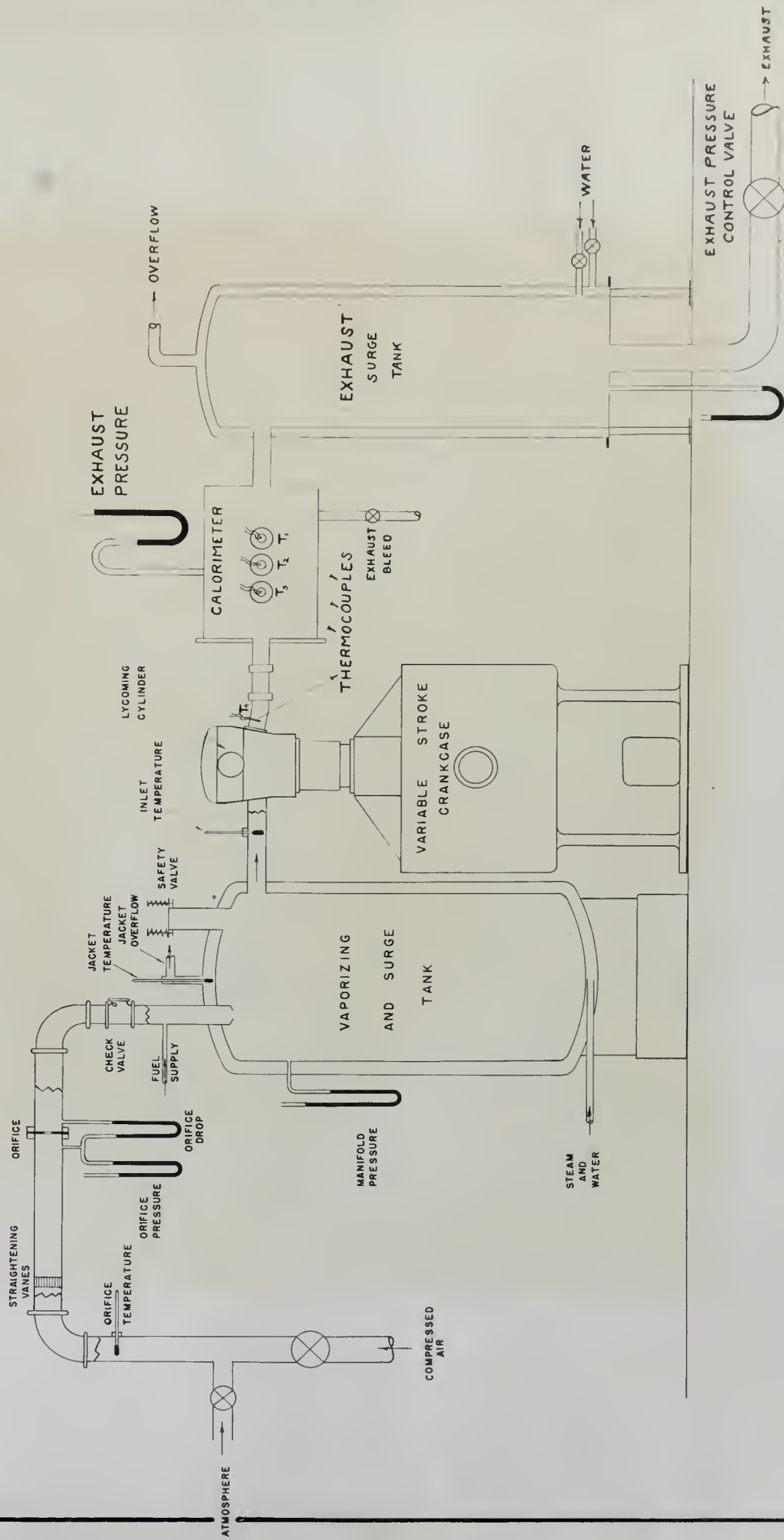


Fig. D

DIAGRAMATIC SKETCH OF ENGINE SET-UP

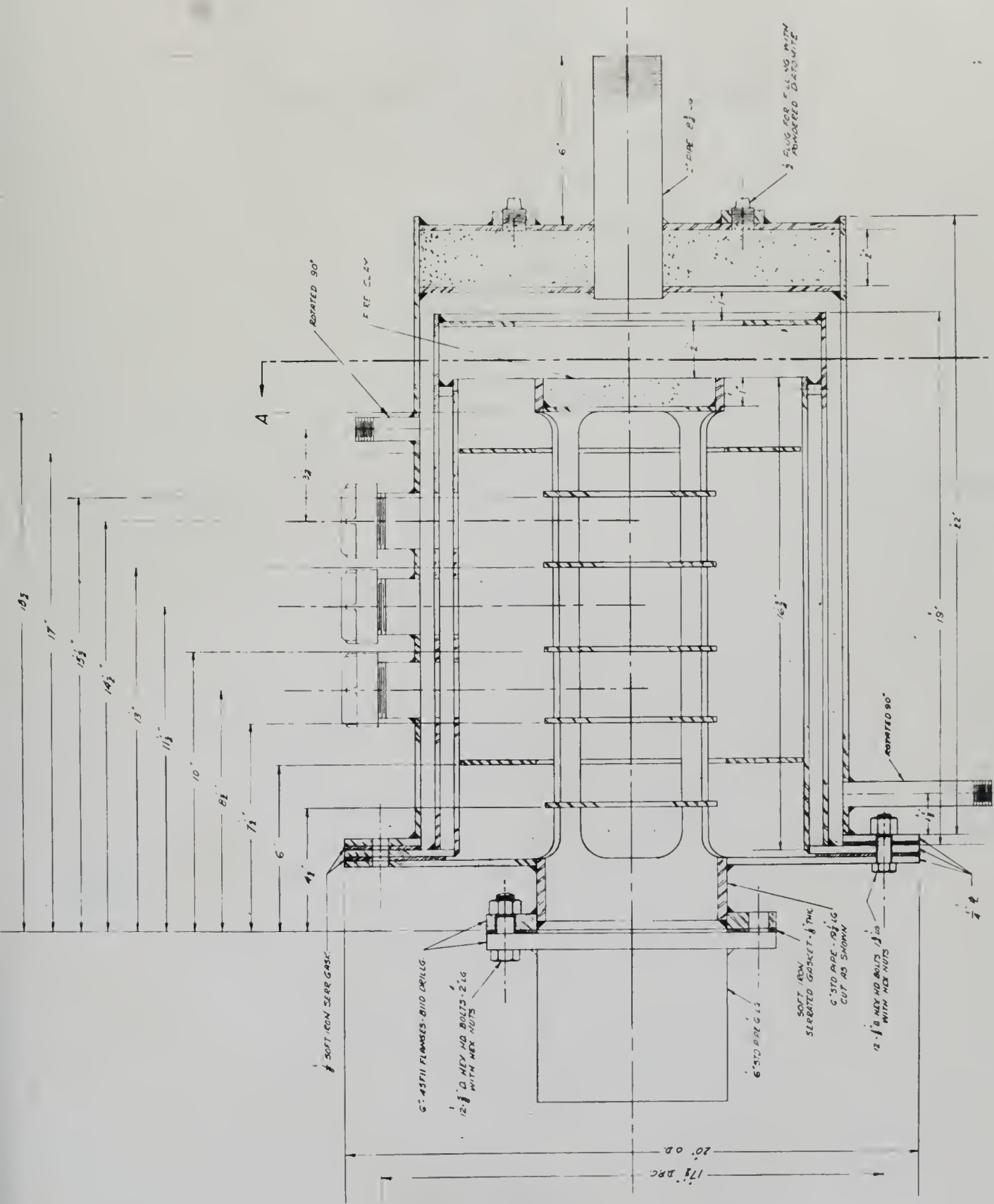


Fig. E
DIAGRAM OF EXHAUST CALORIMETER

VALVE TIMING DIAGRAM

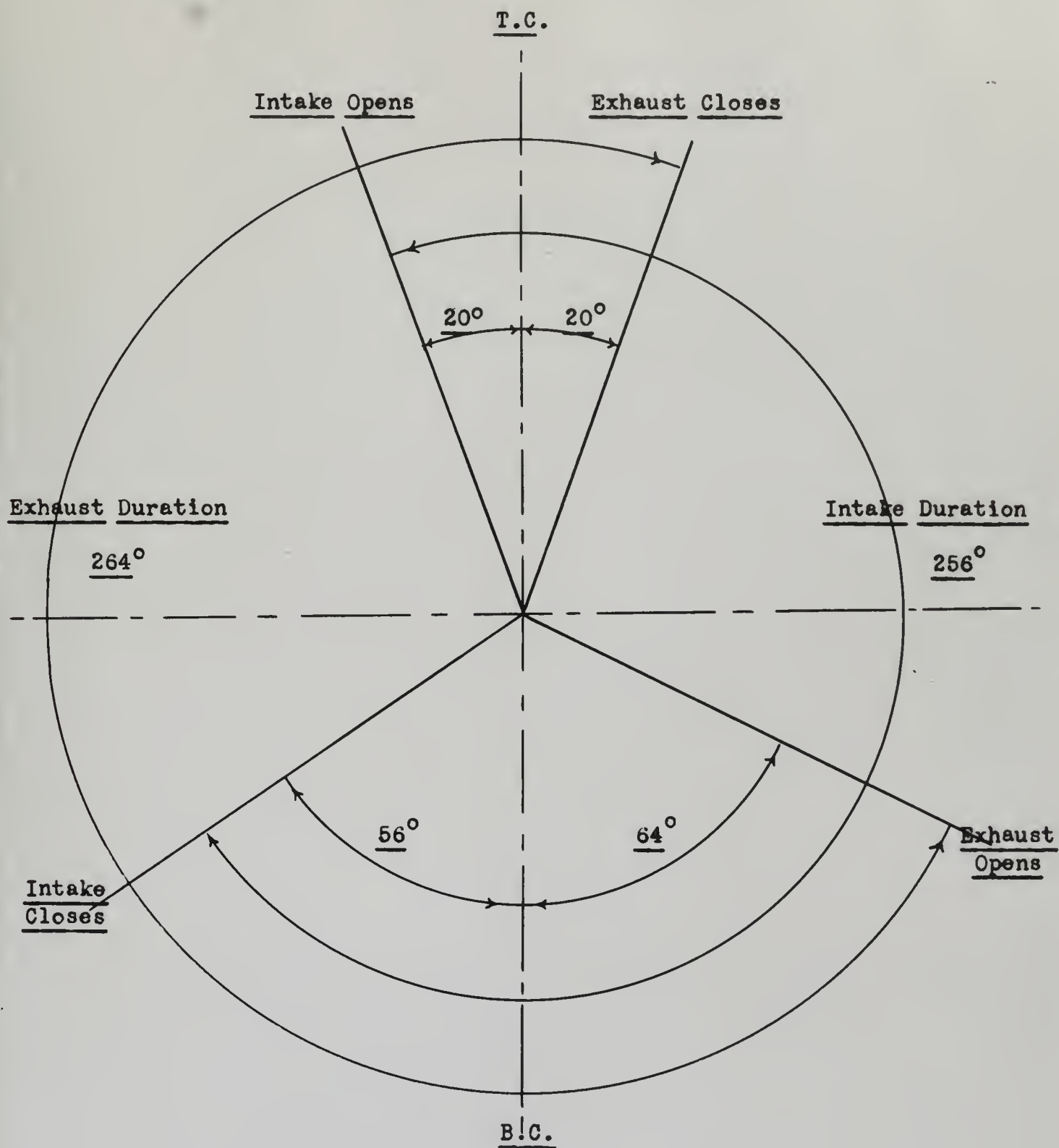


Fig. F

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engine performance and
the availability of
energy in exhaust gases
at cruising conditions.

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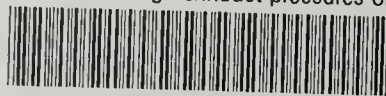
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